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National Aeronautics and
Space Administration

ENERGY EFFICIENT ENGINE
ICLS ENGINE
BEARINGS, DRIVES, AND CONFIGURATION

DETAIL DESIGN REPORT

by

C. L. Broman

GENERAL ELECTRIC COMPANY
Aircraft Engine Business Group

JUNE 1982

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16. Abstract

The detailed design of the forward and aft sumps, the accessory drive system, the lubrication system, and the piping/manifold configuration to be employed in the ICLS engine test of the Energy Efficient Engine is addressed in the report. The design goals for the above components were established based on the requirements of the test cell engine.

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FOREWORD

This report documents technical analysis and design performed by the General Electric Company for the National Aeronautics and Space Administration, Lewis Research Center, under Contract NAS3-20643. The work was performed as part of the Aircraft Energy Efficiency (ACEE) Program, Energy Efficient Engine (E^3) Project. Mr. C. Ciepluch is the NASA Project Manager. Formerly, Mr. W. Hady was the NASA Project Engineer responsible for managing the effort associated with the Bearings, Drives, and Configuration design presented in this report. Presently, Mr. T. Strom is the responsible NASA Project Engineer.

Mr. R.W. Bucy is the Manager of the Energy Efficient Engine Project for the General Electric Company. Mr. J.C. Clark is the Manager responsible for the technical effort relating to Bearings, Drives, and Configuration for the General Electric Energy Efficient Engine.

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1.0 SUMMARY

As part of the contract requirements of the Energy Efficient Engine (E^3), a detailed design review was held on the Bearing Systems, Drives, and Configuration for the Integrated Core Low Spool (ICLS) engine. This review was held at the NASA-Lewis Research Center on April 23, 1981.

The review included analysis of the bearing loads and lives, gear stresses, and structural design of the forward and aft sumps, and the accessory drive system. The schematic of the lubrication system was presented along with a discussion of the secondary air systems surrounding the sumps. The design of the major configuration piping was also addressed.

As a result of this design review, approval was granted by the NASA-Lewis, Energy Efficient Engine Project Manager to proceed with the procurement of the respective hardware which was discussed during the review.

2.0 INTRODUCTION

A major milestone in the development of the General Electric E^3 engine is the ICLS test scheduled for the fourth quarter of 1982. This report describes the design of the bearing systems, drives, and configuration required for the ICLS test. Since many of the drive system components to be used in the core/ICLS test are common to other General Electric engines, the design goals for these components have been established based on the requirements of the test cell engine. Table I shows these program goals compared to the design requirements of the flight propulsion system (FPS) which represents the fully developed engine installed on an aircraft. A detailed description of the E^3 FPS is given in References 1, 2, and 3, and detail design report for the core engine is given in Reference 4.

A cross section of the ICLS engine is shown in Figure 1. Identified in this figure are the forward sump, aft sump (including the LP shaft) and the accessory gearbox. The lube system, secondary airflow system surrounding the sumps, the engine rotor thrust and the configuration piping were also included in this design review. Shown in Table II are the basic design features included in the design of the ICLS engine.

Referring to Figure 1, the forward sump contains the compressor rotor forward thrust bearing, the power takeoff (PTO) gearbox, the fan rotor thrust bearing, and roller bearing.

The PTO provides a drive which connects the engine rotor system to the accessory gearbox (AGB) mounted on the outer engine casing. Engine-required accessories, including drives for two starters, are provided on the AGB. The AGB is designed to be used on both the core and ICLS engine.

The ICLS aft sump contains the intershaft bearing which supports the aft end of the core rotor system through the LP shaft aft bearing.

Both the forward and aft sumps are vented through a central tube which terminates aft of the engine exhaust nozzle. An instrumentation slipring is mounted behind the aft sump and the sump vent air is directed around the slip-ring assembly.

The lube system is typical of other large turbofan engines built by the General Electric Company. Many components used in the E^3 engine are obtained from other engine programs where performance has been proven.

The configuration piping utilizes a substantial amount of core engine hardware but some piping will be unique to the ICLS engine.

Table I. ICLS Engine Basic Design Goals.

	FPS Requirements	ICLS Engine
• Gearbox Gears and Bearings	36,000 hours	2,000 hours
• Number of Starts	40,000	2,000
• Mission Cycles	36,000 cycles	10,000
• Bladeout		
– Safe Shutdown	635×10^3 g-mm (25,000 g-in.) Fan Unbalance (1 Fan Bld. + 30%)	635×10^3 g-mm (25,000 g-in.) (Min. Secondary Damage)
– Continued Operation	- 1 - 1-1/2 HP or LP Turbine Bladeout	Shutdown Situation
	- 2 HP Compressor Bladeout	(Damper Design Includes Continuous Unbalance Consideration)
	- 1 HP or LP Dovetail and Adjacent Bladeout	
• Flight Environment	- Mach No. Versus Altitude Profile	Test Pad Conditions
	- Attitudes	

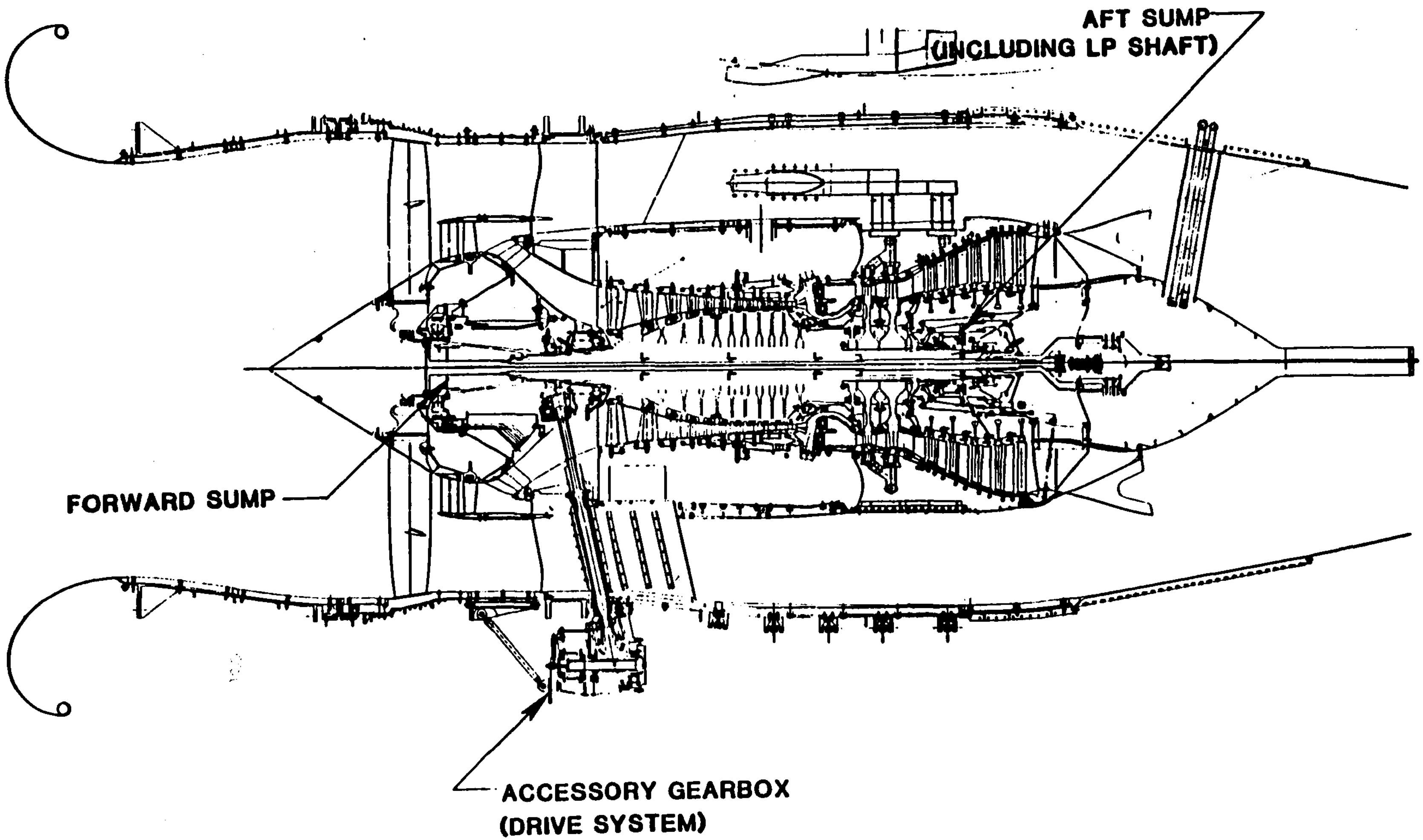


Figure 1. ICLS Engine Cross Section.

Table II. ICLS Engine Basic Design Features.

- Two-Sump Engine
- Five Main Shaft Bearings
  - One Intershaft Bearing
  - Core Bearings Spring Mounted
  - Thrust Bearing Fluid Damped
- Labyrinth Seals
- Center Vent System
- Drive System Commonality Between Core and ICLS
  - Provide for Core or Fan-Mounted AGB
- Major Core/ICLS, FSFT Hardware Commonality

3.0 FORWARD SUMP

The design of the forward sump used in the ICLS engine for the E^3 engine is shown in Figure 2. This sump includes the core thrust bearing (No. 3) which is supported by a machined centering spring housing which is provided with a multishim damper.

The PTO gearbox also located in the forward sump is driven directly from a gear-mounted (on the compressor) stub shaft. The drive shaft from the PTO gearbox to the externally mounted accessory gearbox exits from the forward sump through the bottom strut in the frame.

The forward sump also includes the fan thrust bearing (No. 1) and a fan shaft support roller bearing (No. 2). Located just forward of the No. 2 bearing is the LP speed sensor device used in the engine control system.

The forward sump seals are located just forward of the No. 1 bearing and aft of the core thrust bearing. Labyrinth seals are used which are pressurized with fan discharge air obtained through the leading edge of the core struts and forward of the compressor inlet. Pressurization air is directed to the aft sump by way of the LP shaft. The sump is vented through a "center vent" pipe to a region of low pressure in the exhaust of the engine.

Figure 2 also shows the selection of material for the major components in the forward sump. Marage 250 material is used for the damper spring housing to meet the fatigue strength requirements.

Figure 3 shows the core thrust bearing damper design. Support for the thrust bearing (No. 3) is through the damper spring housing which is secured to the No. 2 bearing housing. The damper spring housing features 34 machined beams which have been configured to obtain an overall spring constant of 525.4 kN/cm (300,000 lb/in). Tapered beams are used to move the high stresses away from the ends where stress concentrations occur. The maximum stress is below the endurance limit of the Marage 250 material when deflected ± 0.508 mm (0.02 in.).

The total damper shim clearance is 1.173 to 1.326 mm (0.0462 to 0.0522 in.), but a mechanical stop is provided at 0.508 mm (0.020 in.) to limit the overall radial excursion of the compressor rotor.

The damper, located radially outward from the thrust bearing, is a five-shim, end-sealed design. Sealing is accomplished by piston rings at each end of the damper assembly, sealing radially between the thrust bearing housing and the stationary structure. Oil is supplied to the damper from a separate manifold and is distributed internally through six equally spaced holes at the outer periphery of the damper. Six check valves are provided to prevent backflow of high pressure oil.

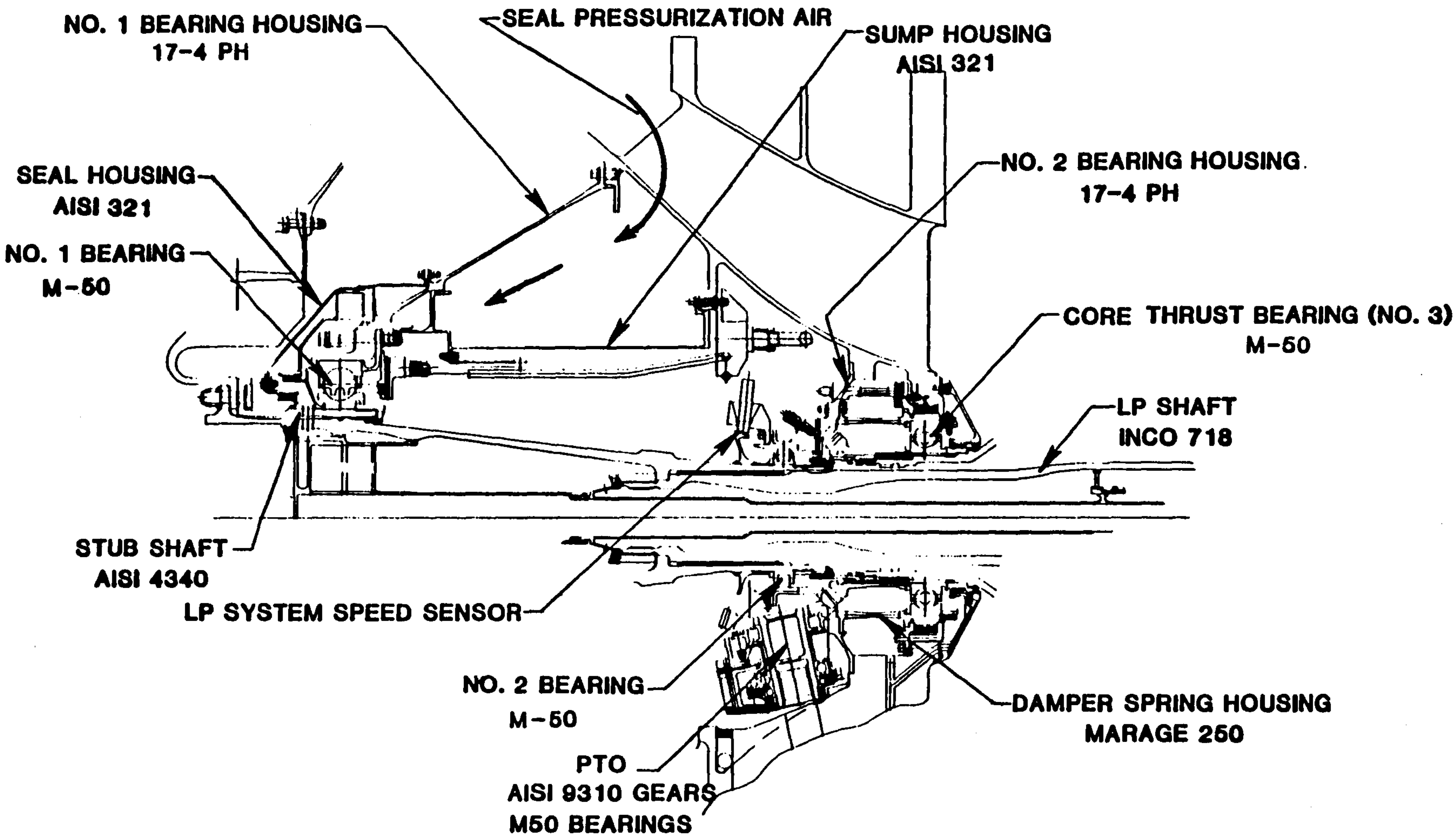


Figure 2. Forward Sump.

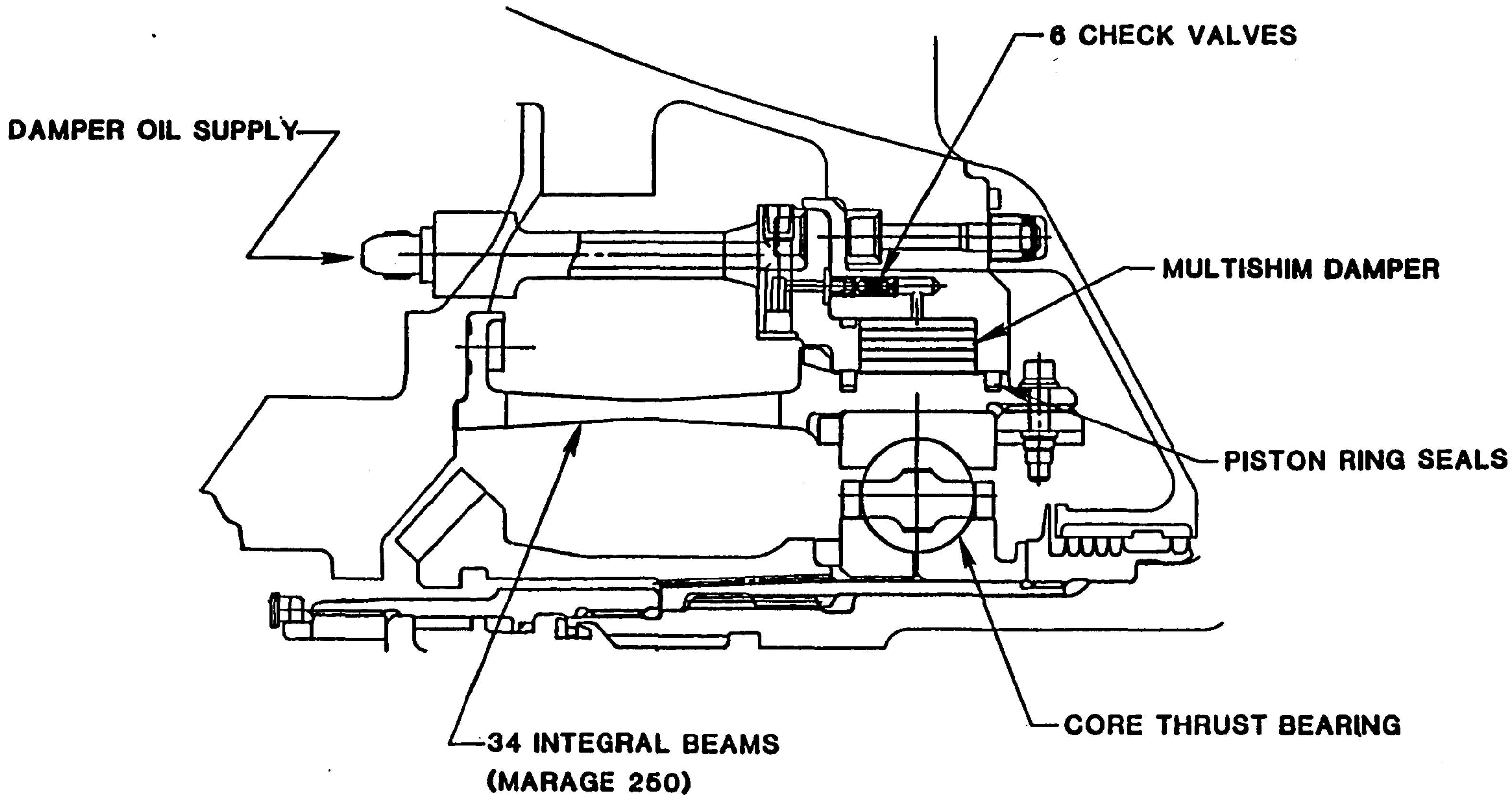


Figure 3. Forward Sump Thrust Bearing Damper Design.

Meehanite, a gray cast iron which has been shown to have excellent wear characteristics, is being used for the piston rings. The piston ring cross section is 3.81 mm (0.150 in.) and 3.302 mm (0.13 in.) thick. These proportions have been selected to obtain a moment-balanced design when the damper assembly has been deflected 0.249 mm (0.0098 in.) which is the expected deflection due to 38,100 g-mm (1500 g-in.) of high pressure turbine rotor unbalance. The expected damper oil pressure at this eccentricity has been calculated at 3447 to 8273 KPa (500 to 1200 psi) depending on the viscosity characteristics of the oil circulating within the damper.

3.1 CORE THRUST BEARING DESIGN

Figure 4 shows the core thrust along with information pertinent to its design. The maximum DN [bearing bore (mm) x shaft rpm] is 2.16×10^6 which is about 8% higher than current commercial engine experience. This, however, is not expected to be a problem. The bearing is underrace cooled and lubricated through 24 slots in the forward inner ring. The bearing also incorporates antirotation tangs to prevent outer race rotation.

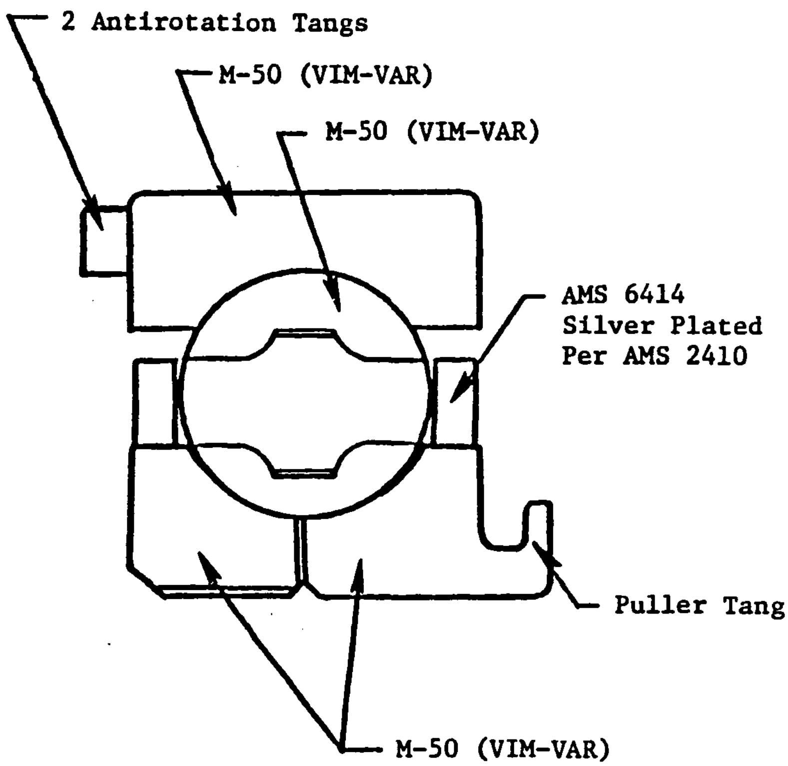
The cubic mean load (CML) for the bearing has been determined by a rotor thrust analysis of a typical commercial flight mission cycle. The calculated L_{10} life, utilizing a multiplying factor established from experimental data on similar bearings, indicates 36,000 hours can be met. Figure 5 shows the calculated relationship of thrust load versus L_{10} life. The L_{10} life should be at least 8400 hours based on the maximum expected thrust load at takeoff conditions.

Studies have been made to determine the bearing fits and clearances as a function of engine speed. Results of this study are shown in Figure 6. The operating contact angle for a minimum and maximum fit condition is expected to be between 22.5° and 27.2° which is within the design requirement for a bearing of this size and operating speed. The fit between the bearing inner race and the shafting reduces as the engine speed increases, but it should still be 0.0076 mm (0.0003 in.) at maximum speed to prevent movement of bearing inner race relative to its shafting.

A component test is to be run to confirm the lube supply system to the bearing. The test rig, shown in Figure 7, includes the core thrust bearing and closely simulates the engine lube supply system. Various thrust loads at idle, intermediate, and maximum speeds will be run with varying lube flows to determine the operating characteristics of the bearing.

3.2 FORWARD SUMP LOW PRESSURE SYSTEM BEARING DESIGN

Figure 8 shows the basic configuration of the No. 1 and 2 bearings. The DN values of these bearings are relatively low and well within current technology. Both of these bearings are jet lubricated with two jets for each bearing. On the No. 2 bearing, one jet is directed between the cage ID and



Bore Diameter,	162.6 mm	(6.4002 in.)
Mean Diameter,	203.2 mm	(8.0000 in.)
Outside Diameter,	243.5 mm	(9.5877 in.)
Element Size,	27 mm	(1.0625 in.)
Number of Elements		20
Maximum Speed		13,300 rpm
DN x 10^6		2.16
Load (cml)	13,833 N	(3110 lb)
L_{10} Life (hr)		36,000

Figure 4. Forward Sump Core Thrust Bearing Design Information.

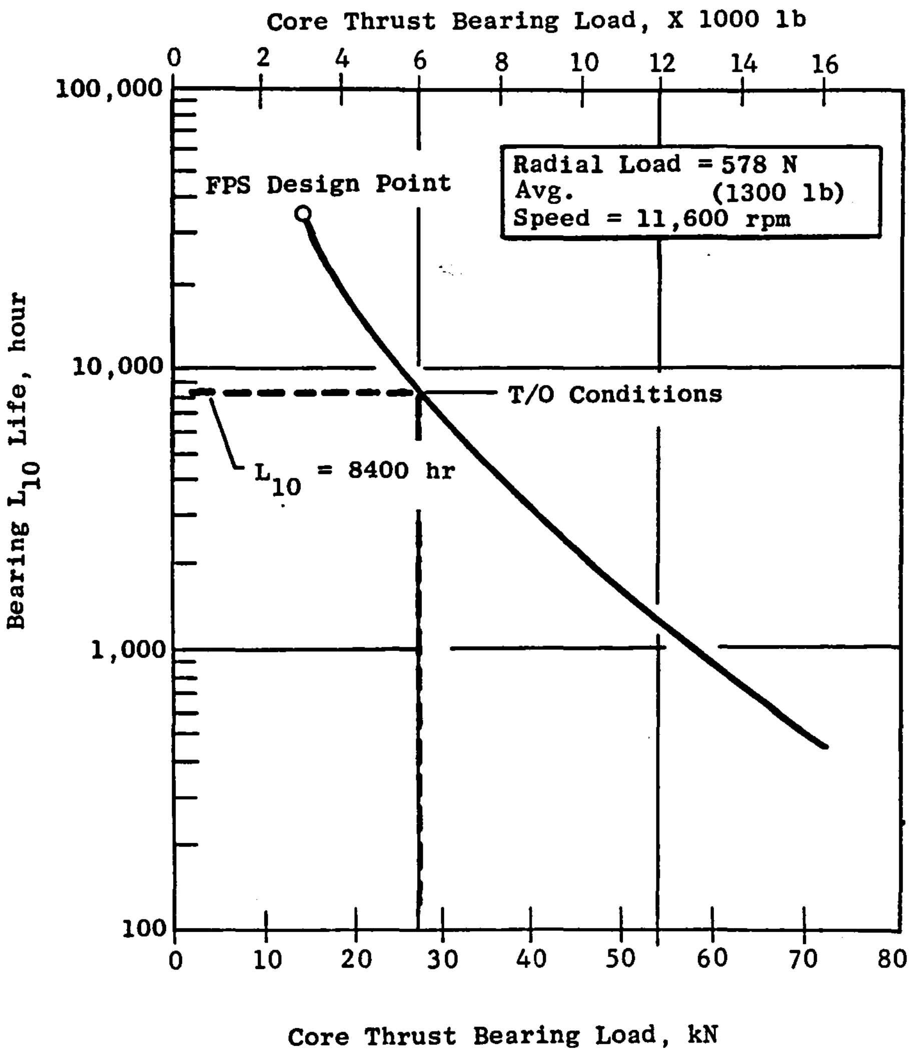


Figure 5. Forward Sump Core Thrust Bearing L_{10} Versus Thrust Load.

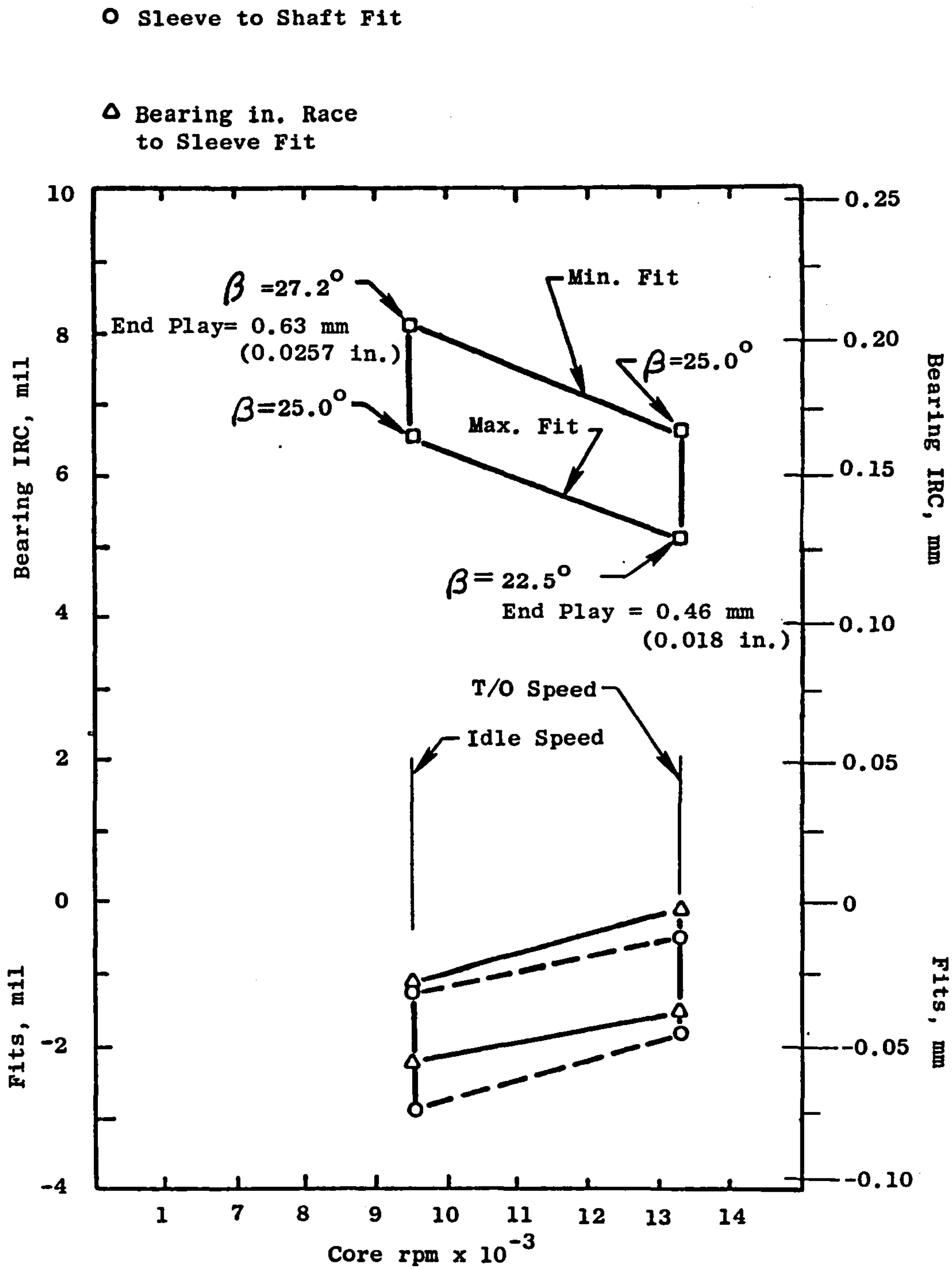


Figure 6. Forward Sump Core Thrust Bearing Fits and Clearances.

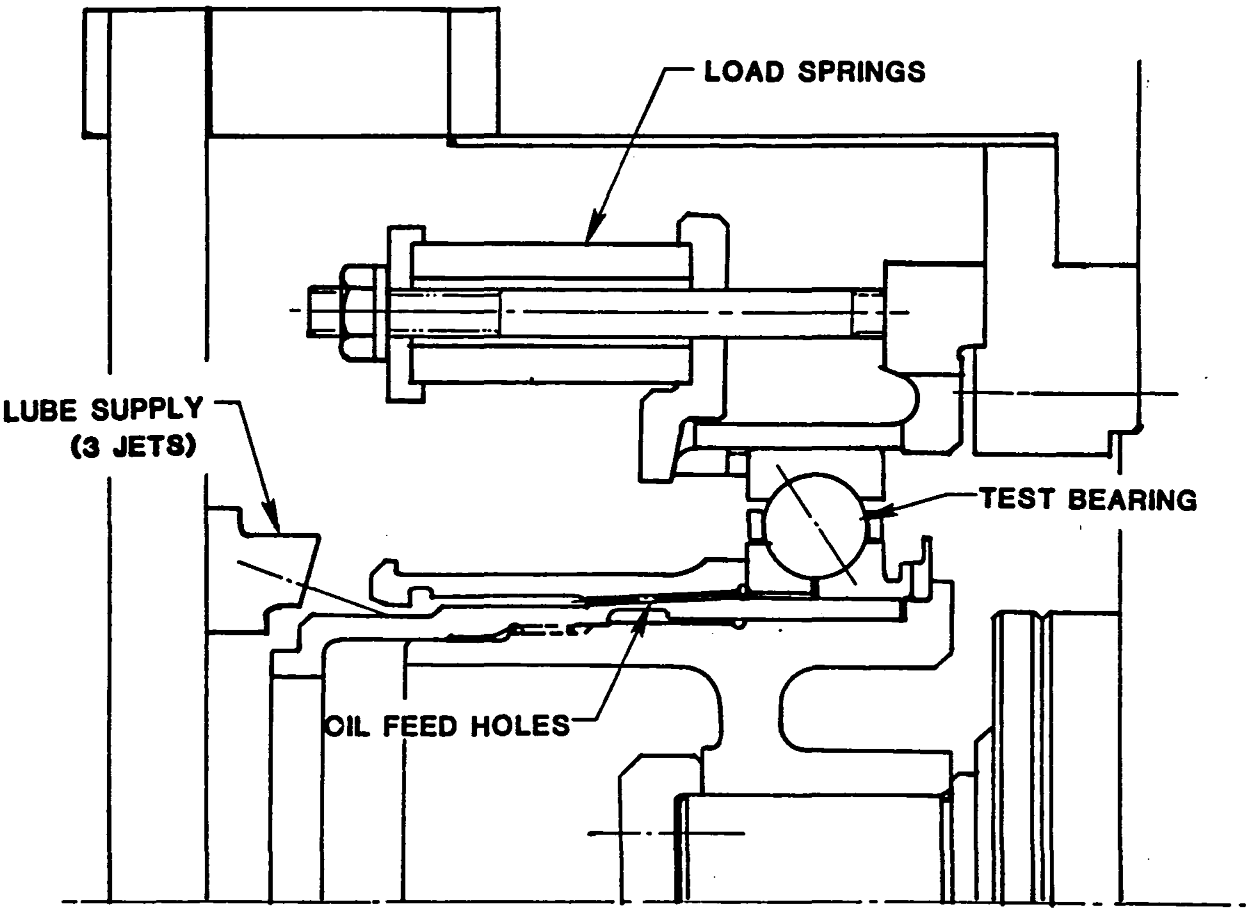
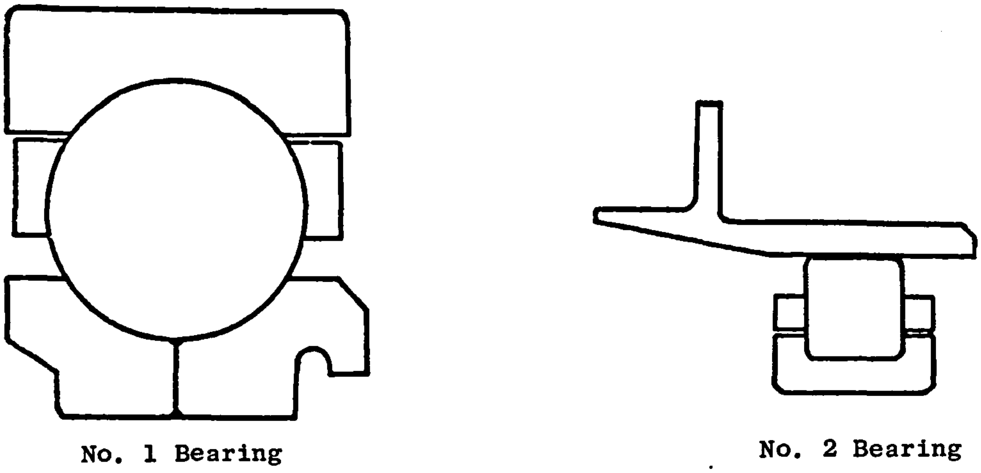


Figure 7. Thrust Bearing Test Rig.



	No. 1	No. 2
Bore Diameter	274.3 mm (10.800 in.)	134.1 mm (5.280 in.)
Mean Diameter	336.5 mm (13.250 in.)	159.8 mm (6.290 in.)
Outside Diameter	392.2 mm (15.600 in.)	185.4 mm (7.300 in.)
Element Size	41.3 mm (1.625 in.)	15x15 mm (0.59x0.59 in.)
Number of Elements	22	26
Max. Speed (ICLS)	3600 rpm	3600 rpm
$DN\times10^6$	0.99	0.48

Figure 8. Forward Sump, Low Pressure System Bearing Design.

the bearing inner ring shoulder and the other is directed between the ID of the outer ring and the cage OD. A large lead-in chamber is provided in the outer ring to facilitate ease of assembly. The same materials are used as the core thrust bearing shown in Figure 4.

The No. 1 bearing fits and clearances are shown in Figure 9. To minimize fretting, the outer and inner rings are assembled tight to their respective housing or shaft. The fit of the inner ring is 0.096 to 0.127 mm (3.8 to 5.0 mils) tight and the outer ring fit is 0.013 to 0.081 mm (0.5 to 3.2 mils) tight. The fits remain tight over the speed range. Operating contact angle, also shown in Figure 9, is according to General Electric design practices.

Figure 10 shows the No. 1 bearing L_{10} life as a function of bearing thrust load. A radial load of 1000 lb is combined with the thrust load to calculate the L_{10} life. The design life for the FPS engine system is to be $\geq$36,000 hours based on a CML calculated from typical flight operating conditions. At ICLS test conditions at takeoff power, the life of the bearing is $\geq$3300 hours.

Shown in Figure 11 are the expected fits and clearances for the No. 2 bearing. Shown superimposed on the bearing IRC is the effect of the shaft torque. This is caused by the separating load of the LP shaft spline located radially inward from the No. 2 bearing; this effect tends to reduce the bearing IRC by 0.076 mm (3 mils) at T/O conditions. There is almost no affect at idle. Also shown in this figure is the bearing inner ring to shaft fit which remains tight throughout the operating range.

3.3 SUPPORT STRUCTURE DESIGN ANALYSIS

Figure 12 shows the results of the stress analysis of the No. 1 bearing housing, the forward stub shaft assembly, and the No. 2 bearing housing, respectively. Shown also in these figures are the selection of materials and other information pertinent to the design.

A stress analysis was conducted on the No. 1 bearing housing (Figure 12) subjected to radial loads due to high unbalance and thrust loads due to engine takeoff power settings. The highest stresses occurs at the smaller end of the conical section where a series of holes penetrate the wall. Stress in this area, including the effects of stress concentrations, is 607 MPa (88 psi) which is below the material allowable stress of 807.3 MPa (117 ksi).

The No. 1 bearing outer race is clamped axially by a contoured plate preloaded during assembly to 133,400 to 222,440 N (30,000 to 50,000 lb) which is greater than the expected maximum bearing thrust load of 53,400 N (12,000 lb) insuring that the bearing will be seated at all times. The clamp plate should withstand loads up to 667,200 N (150,000 lb) without yielding.

The results of the stress analysis of the forward stub shaft assembly is shown in Figure 13. Stresses were calculated due to high unbalance loads and

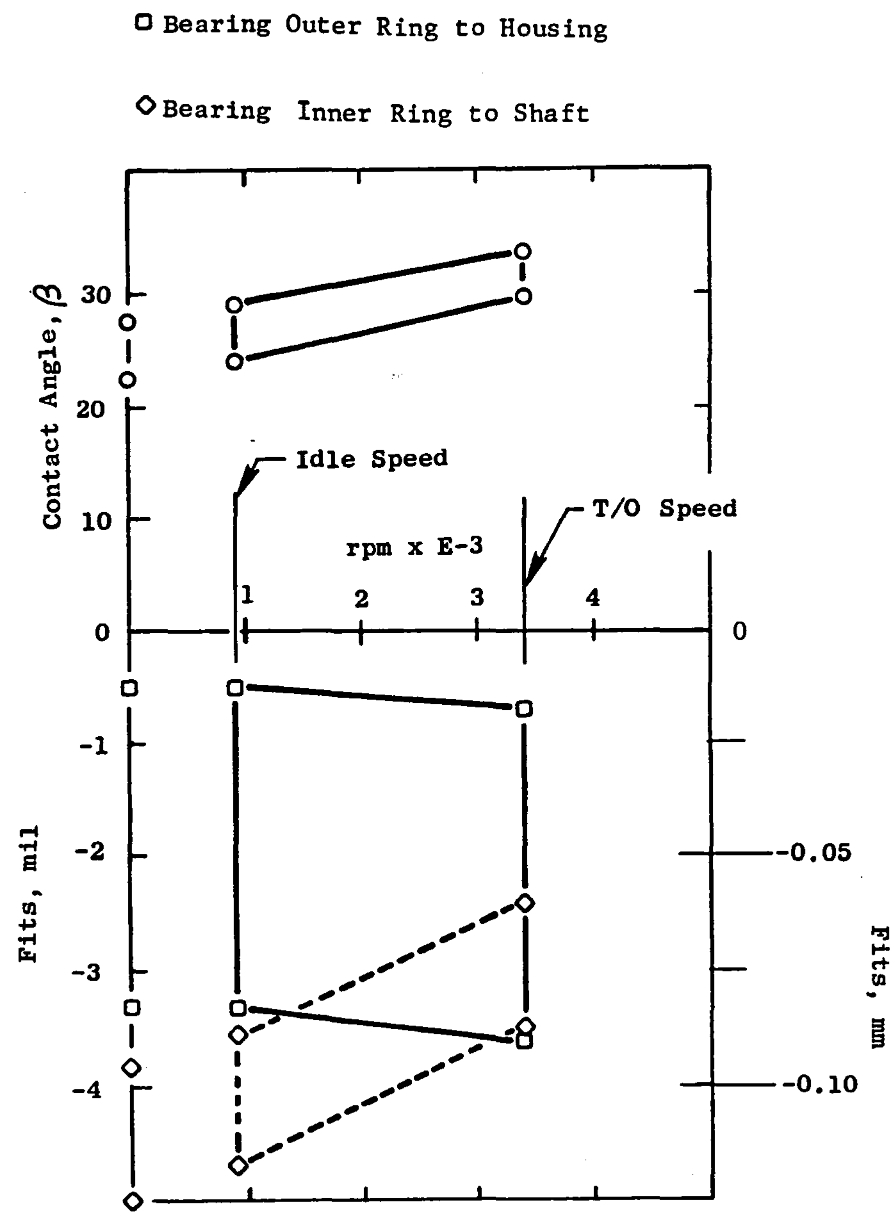


Figure 9. Forward Sump No. 1 Bearing Fits and Clearances.

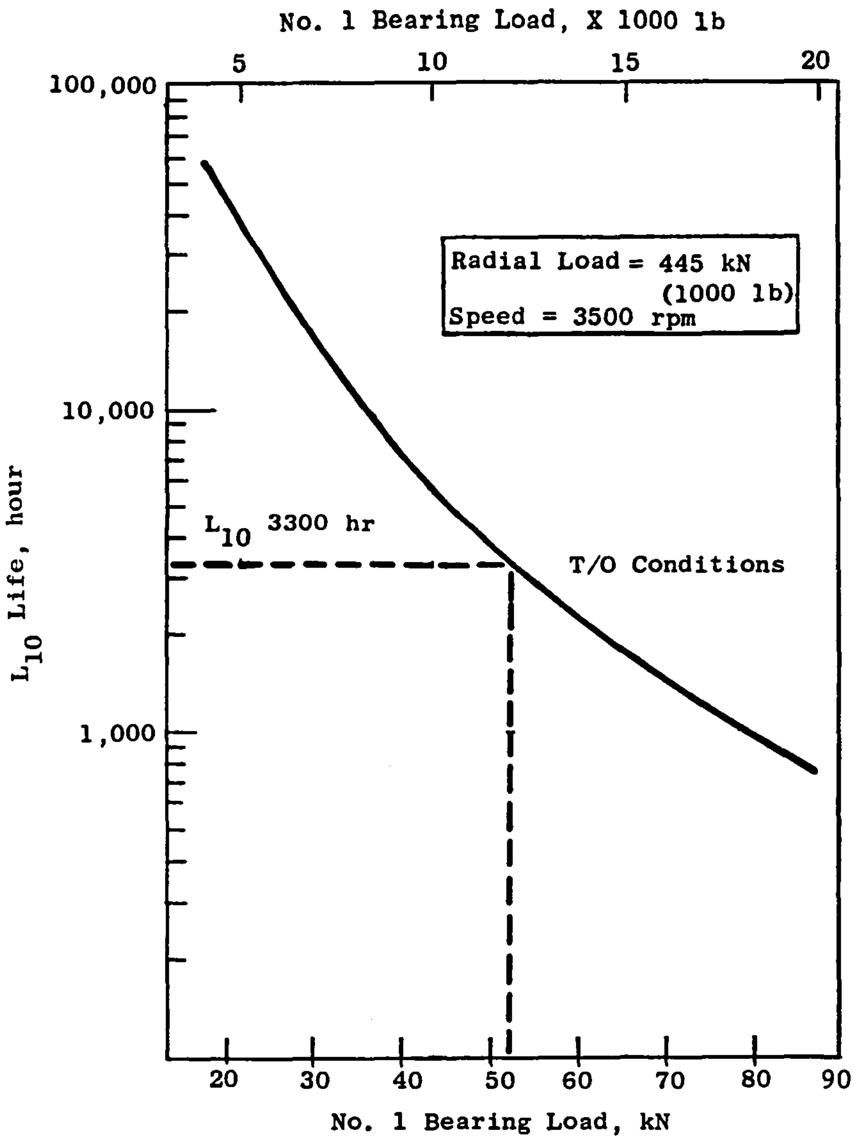


Figure 10. Forward Sump No. 1 Bearing L_{10} Versus Thrust Load.

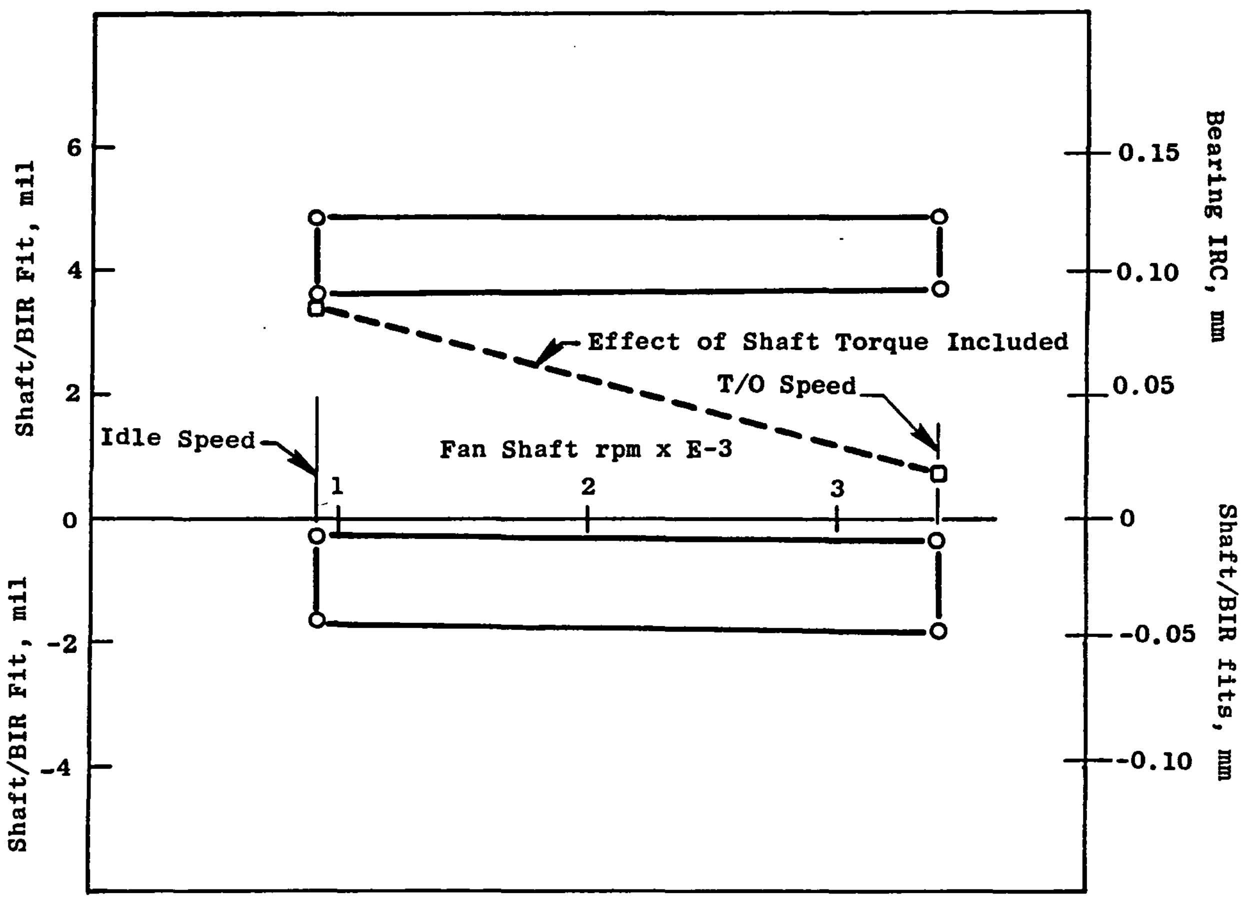


Figure 11. Forward Sump No. 2 Bearing Fits and Clearances.

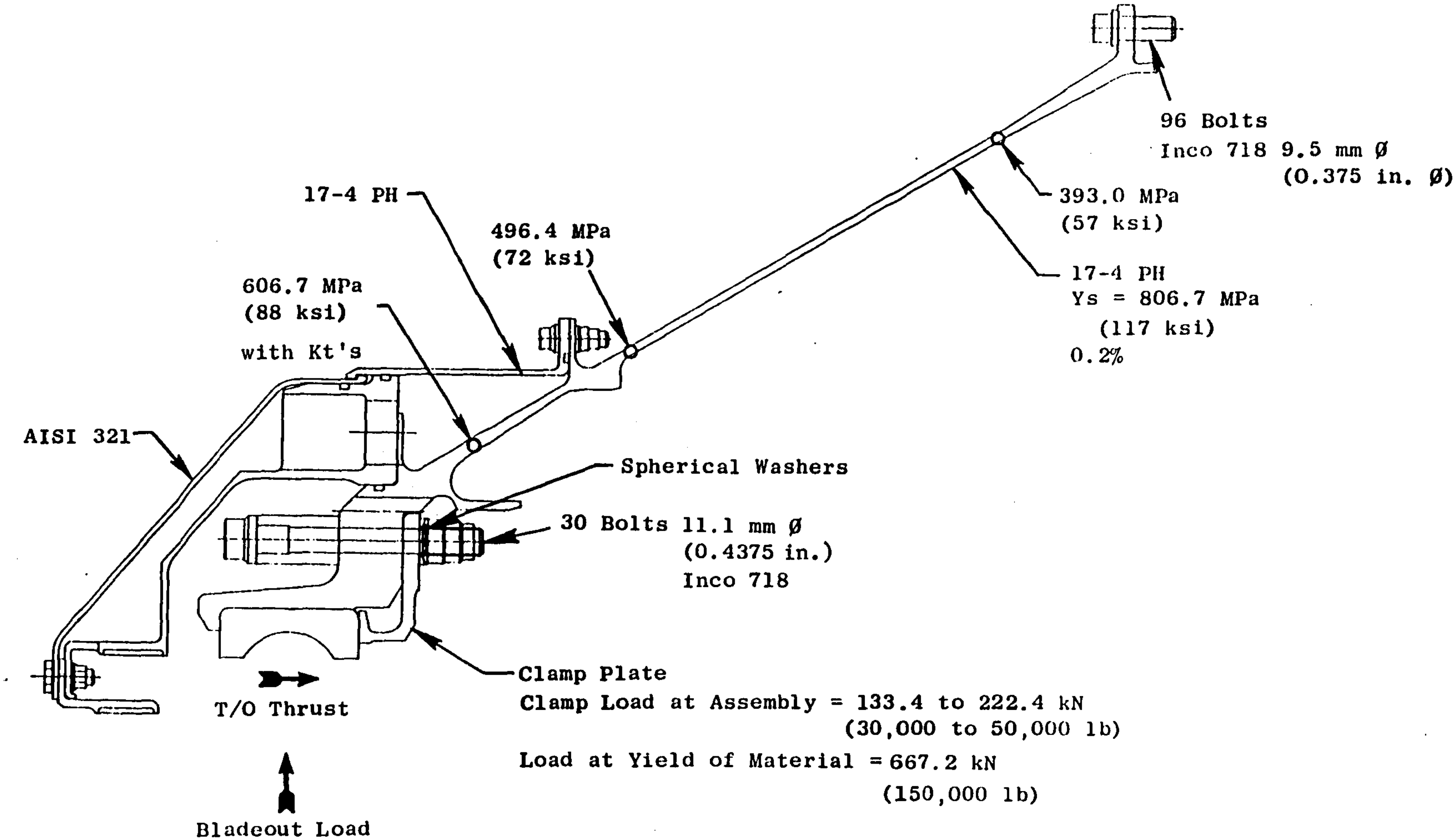
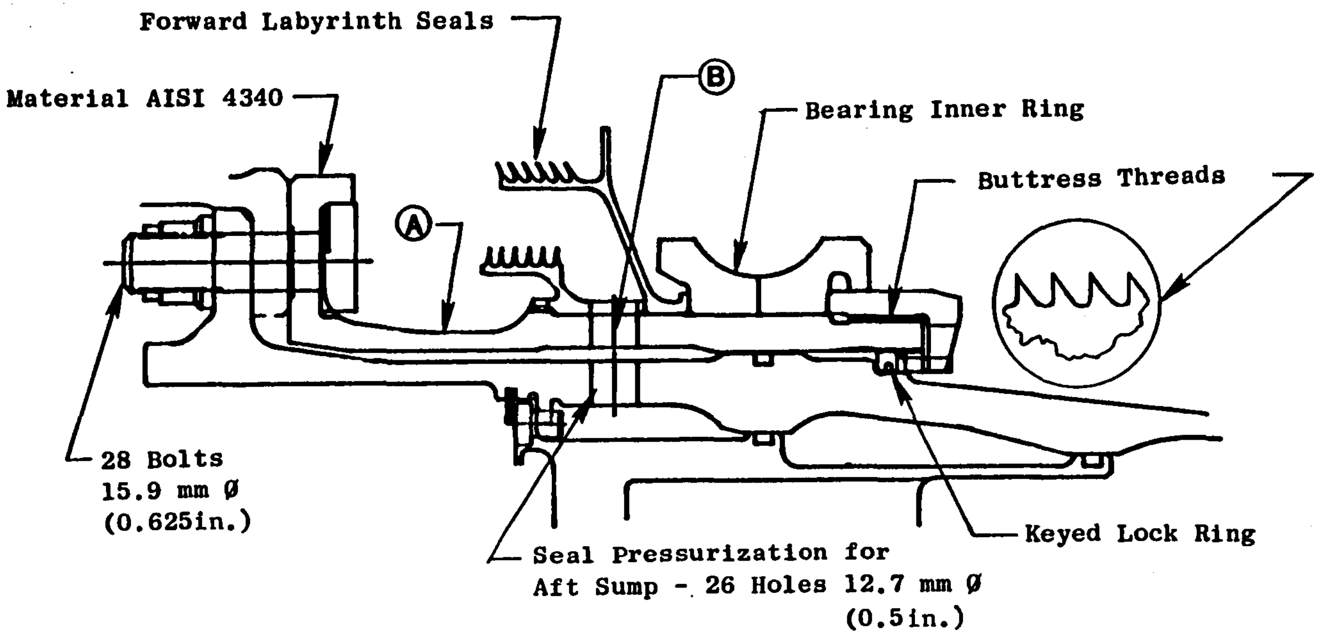


Figure 12. Forward Sump No.1 Bearing Housing.



Mode	Stresses (with Kt's)		Material Allowable
	A	B	
Fan Bladeout	475.9 MPa (69 ksi)	665.6 MPa (96.5 ksi)	0.02% Sy = 862.2 MPa (125 ksi)

Figure 13. Forward Sump Forward Stub Shaft Assembly.

all calculated values are below the material allowable stress. At Plane B, the effects of stress concentration are included. A buttress thread is used on the bearing clamp nut to prevent "thread jumping" due to high unbalance loads.

Figure 14 shows the results of the stress analysis of the No. 2 bearing housing. The highest area of stress is in the vicinity of a slot through which the PTO bevel gear will penetrate. With the housing subjected to high fan unbalance loads, the highest stress is 490 MPa (71 ksi), including the effects of stress concentration, which is below the material allowable stress.

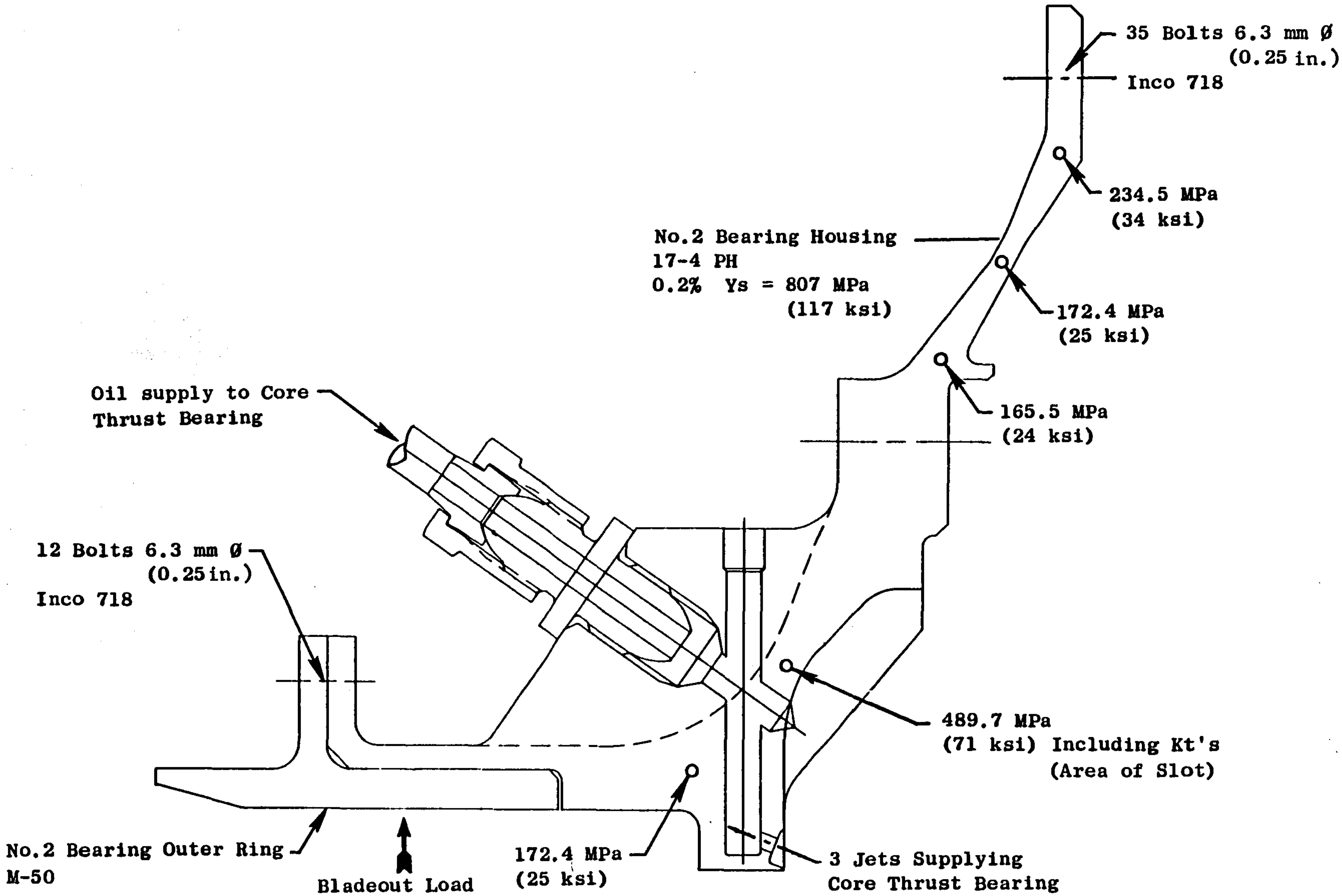


Figure 14. Forward Sump No. 2 Bearing Housing.

4.0 AFT SUMP

The aft sump design configuration is shown in Figure 15. The aft sump includes the No. 4 intershaft bearing and the No. 5 bearing. The intershaft bearing supports the aft end of the high pressure rotor system through a spring housing mounted to the LPT shaft whose bearing is mounted to the No. 5 bearing housing secured to the aft engine frame.

The sump sealing system consists of labyrinth seals forward of the intershaft bearing and aft of the air/oil separator. These seals are pressurized by air from the forward sump area. Outboard of the seal pressurization cavities are cavities which carry the compressor rotor cooling air which exhausts to an area of low pressure in the aft cone area.

The sump is vented to the center vent tube utilizing a dynamic air/oil separator to prevent oil loss out the vent system.

Inco 718 is the primary material used in the aft sump with the exception of the centering spring housing which is Marage 250, the same material used in the forward sump spring housing. The air seals, No. 4 bearing, and spring housing are the same hardware as used in the core engine.

Shown also in Figure 15 is the modification required to the LPT shaft for instrumentation leadout This instrumentation is lead aft to the slipring assembly which is shown in Figure 16. The engine vent system is directed around the slipring assembly and exhausts to ambient pressure aft of areo station A_8. Sump air is vented through a flame arrestor. This part is used on an existing production engine.

Details of oil delivery to the No. 4 and 5 bearing are shown in Figure 17. Both the No. 4 and 5 bearing are supplied through the bore of the LPT shaft; 20% of the total oil is provided to the No. 5 bearing through the bearing inner ring. Holes direct oil to the cage riding surface and the rolling elements. The remaining 80% of the total oil is supplied to the No. 4 bearing through holes in the bearing clamp nut to a stepped sleeve which directs oil through the bearing from the aft side.

4.1 AFT SUMP BEARING DESIGN

The configuration of the aft sump bearings is shown in Figure 18. The material selection for these bearings is the same as the core thrust bearing shown in Figure 4. The design approach used in the No. 4 intershaft bearing is similar to that used on other General Electric engines. Some initial rig testing of a similar bearing running at E^3 speed conditions or equivalent DN indicated no problems should be encountered.

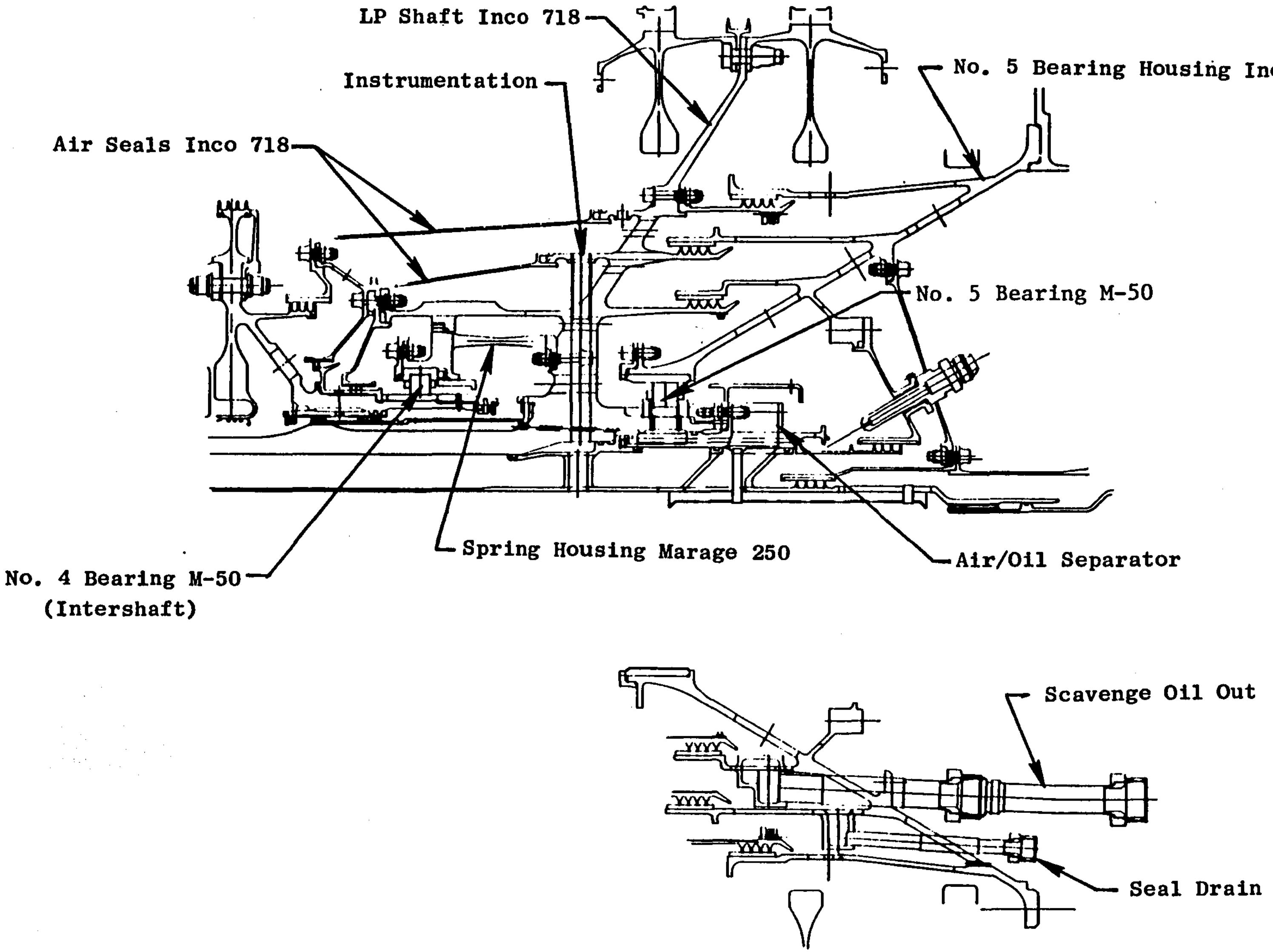


Figure 15. Aft Sump.

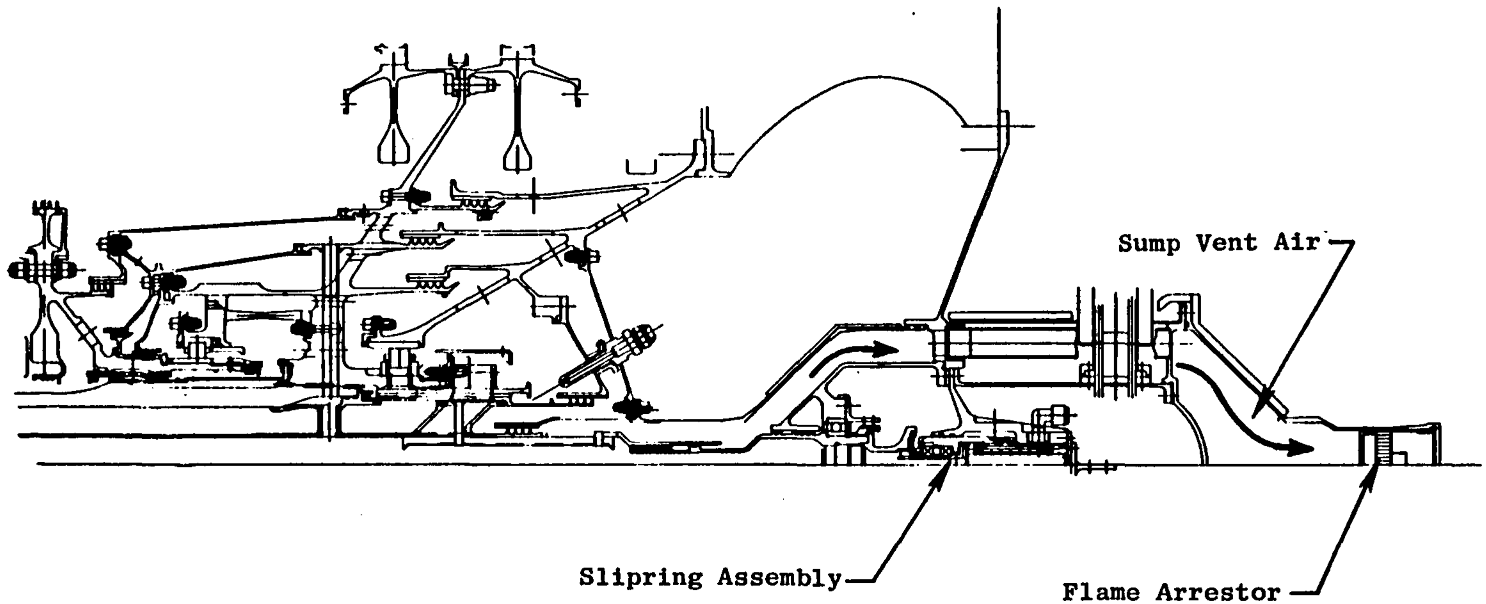


Figure 16. Aft Sump Slipring Configuration.

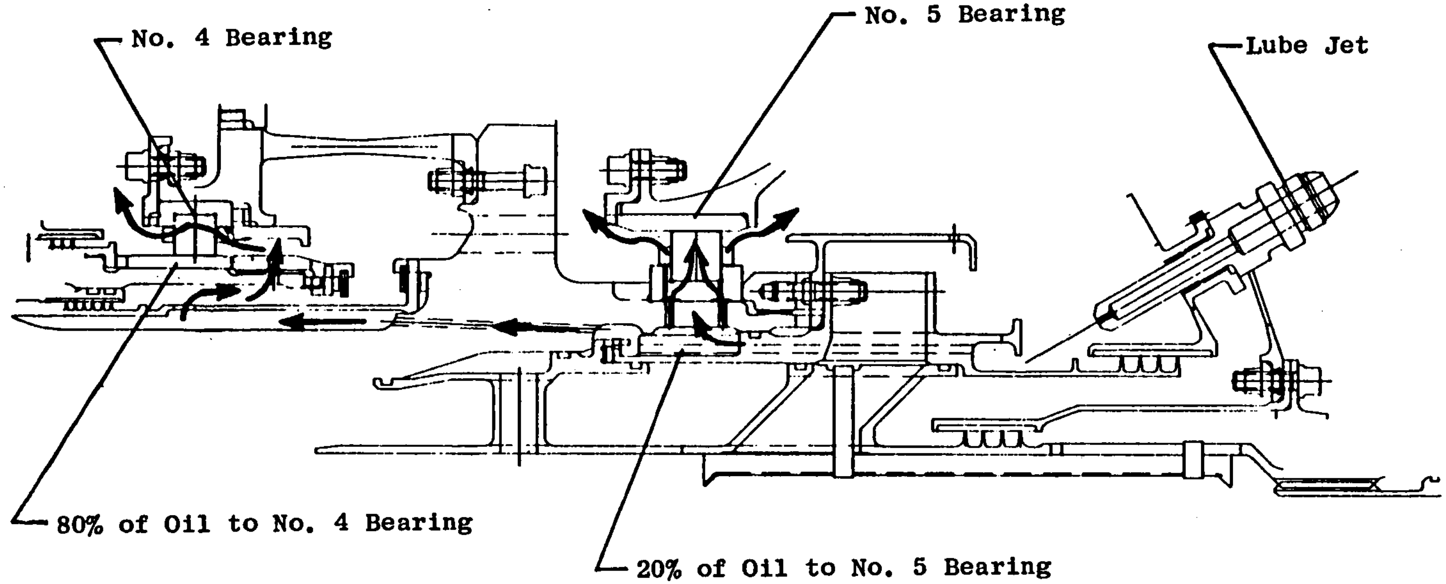
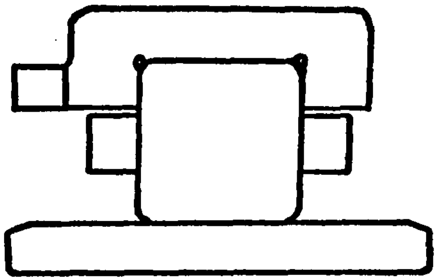
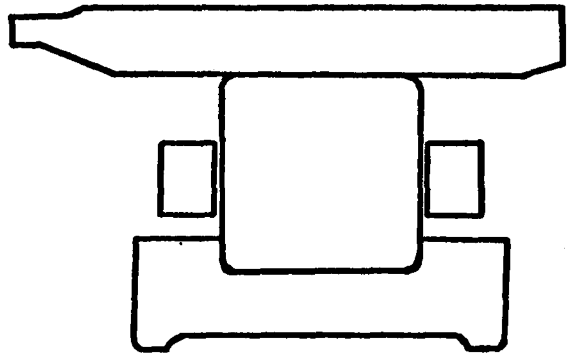


Figure 17. Aft Sump No. 4 (Intershaft) and No. 5 Bearing Lubrication.



No. 4 Intershaft Bearing



No. 5 Bearing

	No. 4		No. 5	
Outside Diameter	219.9 mm	(8.660 in.)	2.113 mm	(8.320 in.)
Bore Diameter	174.1 mm	(6.855 in.)	151.9 mm	(5.980 in.)
Mean Diameter	196.9 mm	(7.751 in.)	182.9 mm	(7.200 in.)
Roller Size	14 x 14 mm	(0.551 x 0.551 in.)	17.75 x 17.75 mm	(0.699 x 0.699 in.)
Number of Rollers		28		26
Max. Speed	9700 (Differential rpm)			3600 rpm
DN x 10^6		1.7		0.55

Figure 18. Aft Sump Bearing Design Information.

The internal fitup of the No. 4 bearing shown in Figure 19 is critical to the successful operation of the bearing. Critical to this design is the need to maintain a preload on the bearing to eliminate roller skidding. The internal clearance of the bearing, the bearing inner race to shaft fit, and bearing outer race to housing fit have been determined to maintain a tight operating condition throughout the operating range. The maximum operating fit is -0.043 mm (-1.7 mils) at T/O speed. At the T/O conditions, the bearing L_{10} life shown in Figure 20, will be 1900 hours which is more than adequate for the ICLS engine.

As shown in Figure 21, the No. 5 bearing which is designed to run with a clearance of 0.061 mm (2.4 mils) at maximum speed is in line with bearings operating at lower DN's.

4.2 SUPPORT STRUCTURE DESIGN ANALYSIS

A stress analysis has been completed on the No. 5 bearing housing, and the results of this analysis are shown in Figures 22 and 23. Figure 22 shows the results of LCF calculation at six critical areas on the housing. Considered in this evaluation were the normal operating loads of the bearing, the thermal gradient along the cone, and the interface loading conditions where the bearing housing mounts to the turbine frame (Point 6). With stress concentrations considered, the LCF design life requirement of > 10,000 cycles is met.

The results of a stress analysis with the housing subjected to high fan unbalance is shown in Figure 23. Stresses due to this high unbalance and the steady-state stresses are combined into an effective stress which is below the allowable yield stress of the material.

All the air seals have been analyzed for critical frequencies. For the rotating and stationary seals, the critical frequencies have been maintained at least 20% above the maximum operating speed. Frequency interactions have been checked between the rotating and stationary seal and there are no interactions up to n = 8 which meets General Electric design guides. Ring dampers have been applied where needed to provide vibration control.

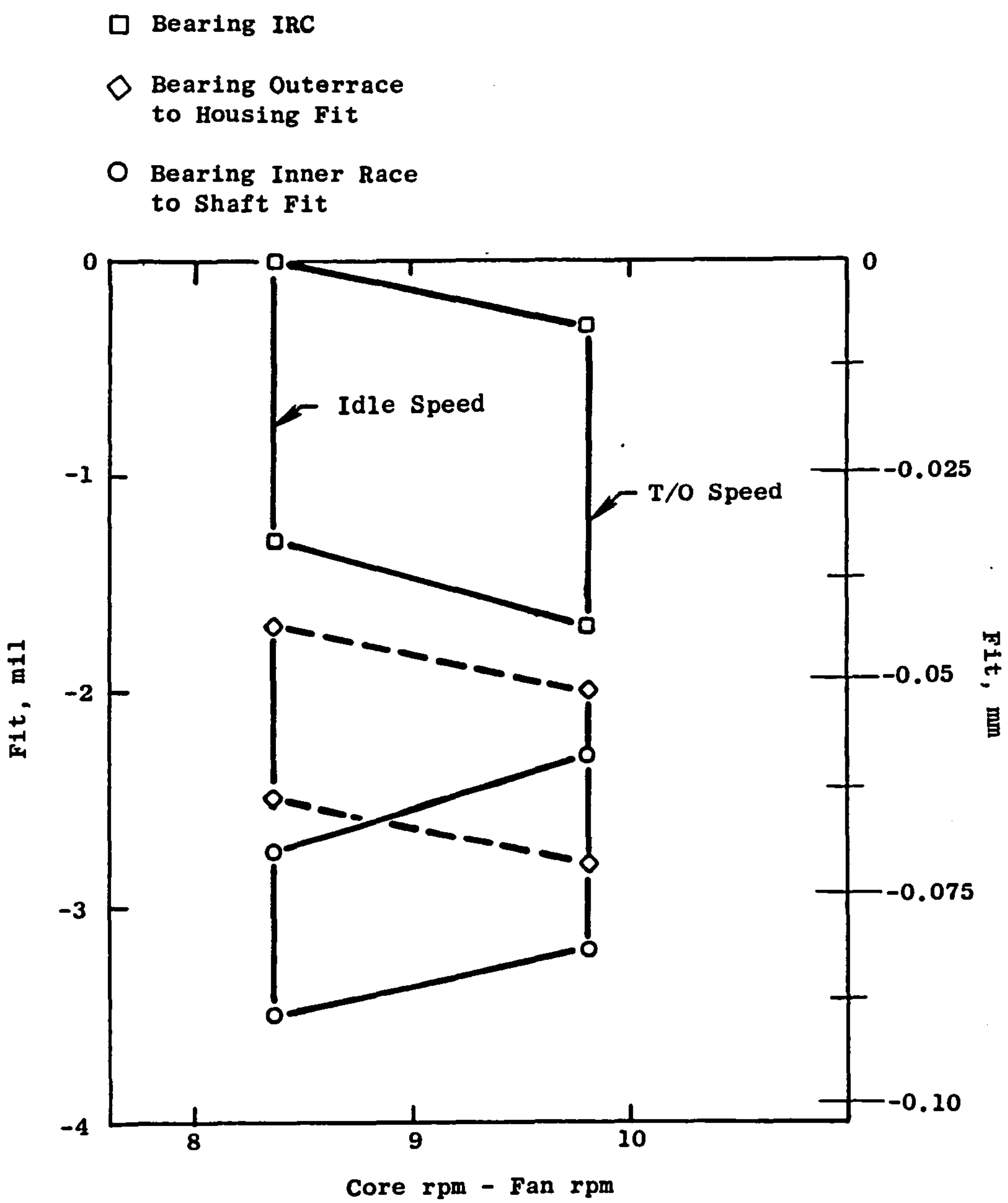


Figure 19. Aft Sump No. 4 Bearing Fits and Clearances.

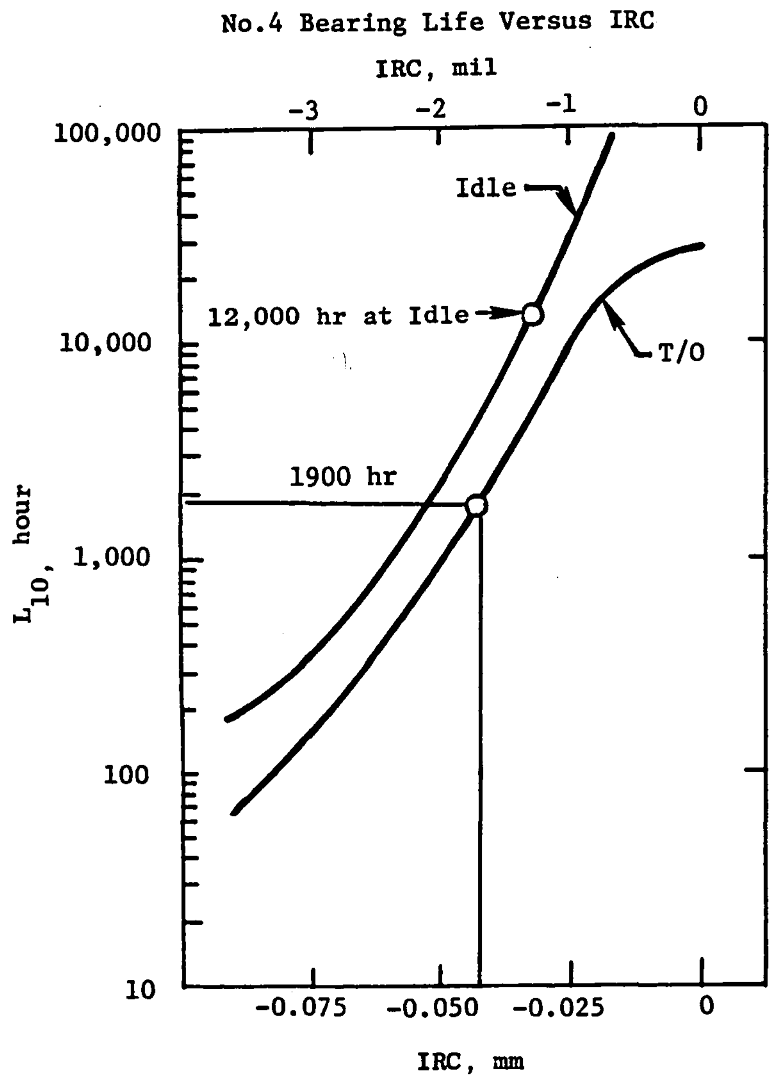


Figure 20. Aft Sump No. 4 Bearing L_{10} Life Versus IRC.

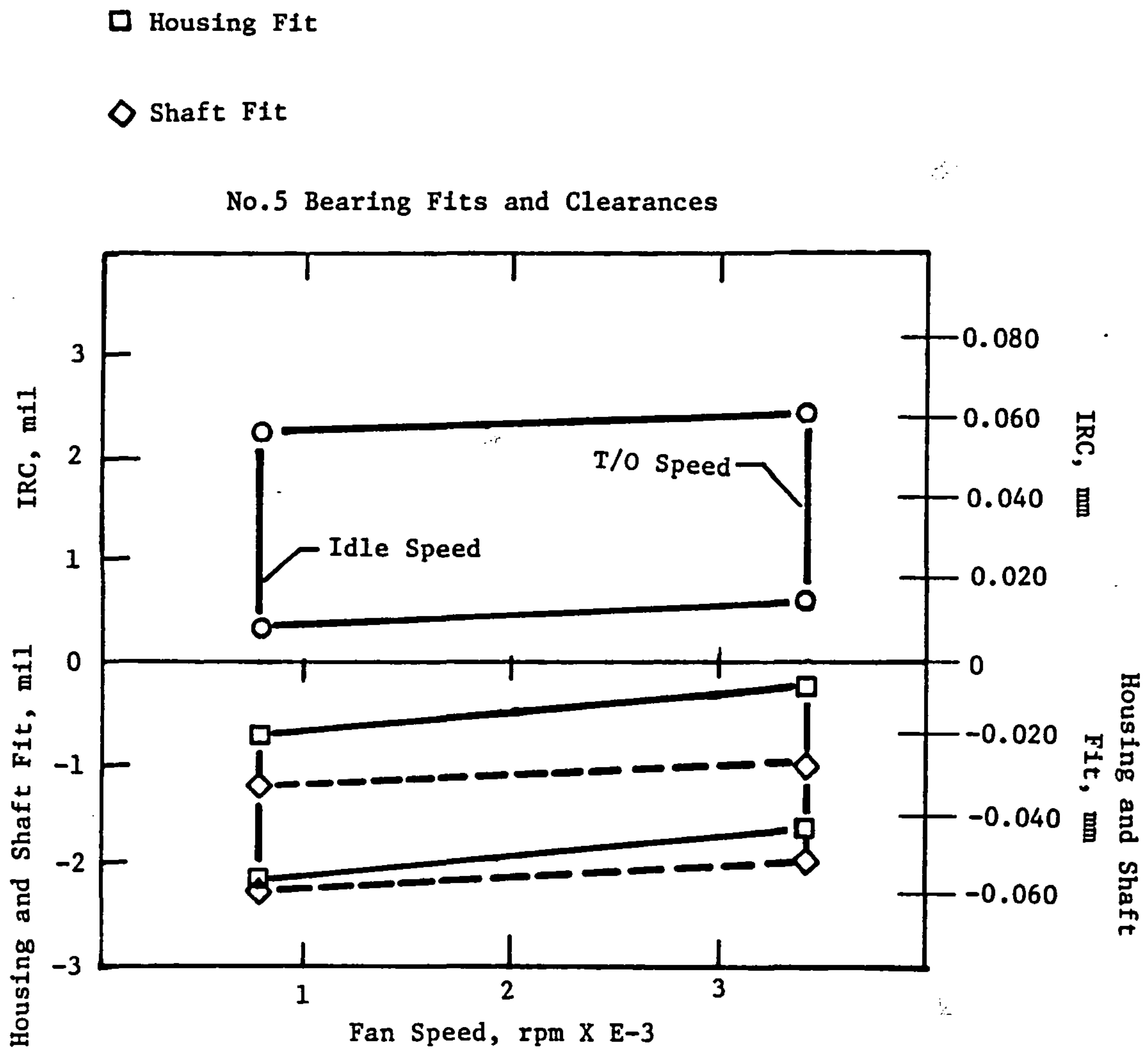
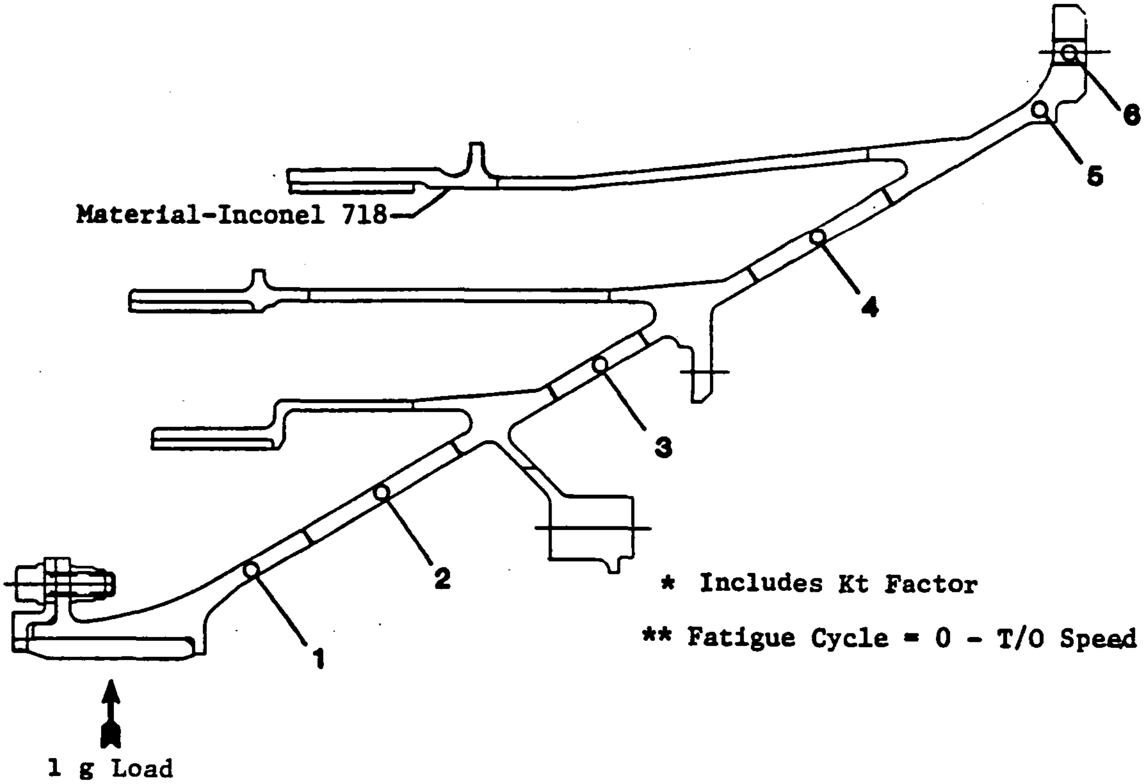


Figure 21. Aft Sump No. 5 Bearing Fits and Clearances.

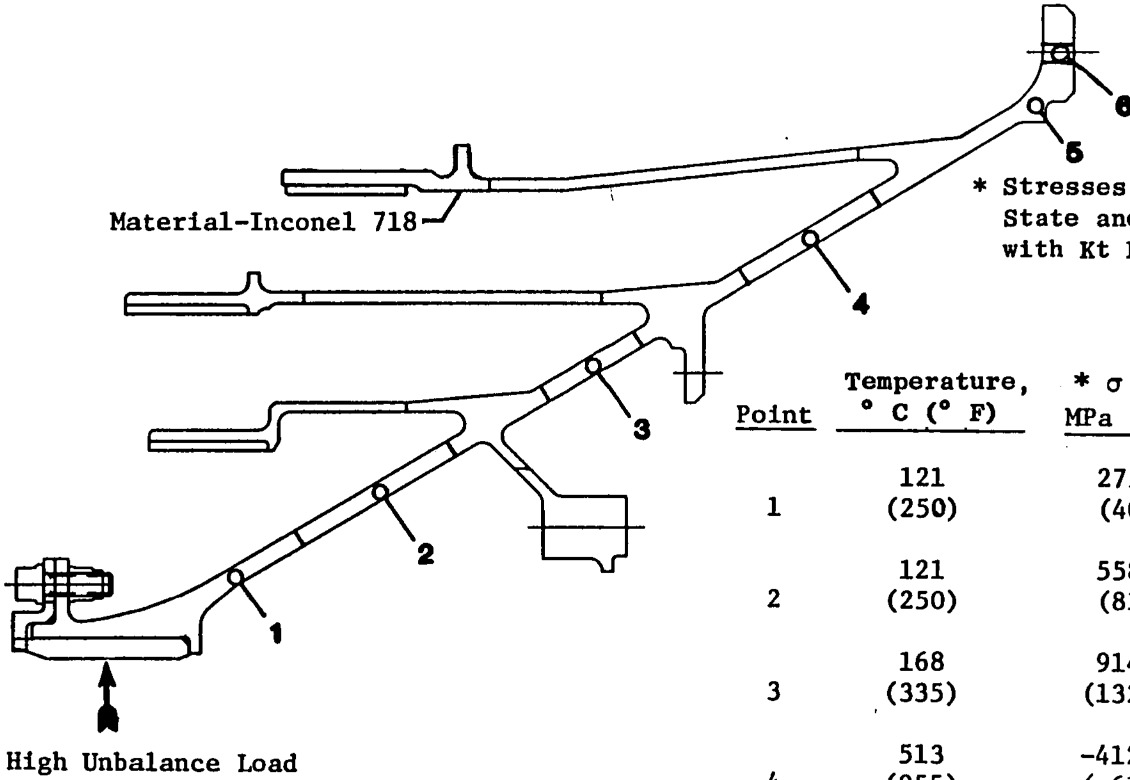


* Includes Kt Factor

** Fatigue Cycle = 0 - T/O Speed - 0

Point	Temperature, °C (°F)	σ Max. (*) MPa (ksi)	σ ss = σ Alt = $\frac{\sigma \text{ Max.}}{2}$ MPa (ksi)	(**) LCF Cycles for A = 1 (-3σ)
1	121 (250)	20.7 (3)	10.3 (1.5)	>10 E5
2	121 (250)	41.4 (6)	20.7 (3)	
3	168 (335)	579.1 (84)	289.2 (42)	
4	513 (955)	599.8 (87)	303.3 (44)	
5	546 (1015)	551.6 (80)	275.8 (40)	
6	546 (1015)	641.2 (93)	320.6 (46.5)	

Figure 22. Aft Sump No. 5 Bearing Support Stress Summary.



* Stresses Reflect the Combination of State and Bladeout Induced Stress with Kt Factors.

Point	Temperature, ° C (° F)	* σ Max. MPa (psi)	0.2% Yield Stress (-3σ), MPa (psi)	
1	121 (250)	275.8 (40,000)	999.7	(145,000)
2	121 (250)	558.5 (81,000)	999.7	(145,000)
3	168 (335)	914.2 (132,600)	999.7	(145,000)
4	513 (955)	-412.9 (-61,200)	868.7	(126,000)
5	546 (1015)	-619.1 (-89,800)	868.7	(126,000)
6	546 (1015)	712.9 (1C3,400)	868.7	(126,000)

Figure 23. Aft Sump No. 5 Bearing Support High Unbalance Stress Summary.

5.0 LOW PRESSURE TURBINE SHAFT

Torque transfer from the low pressure turbine to the forward fan is accomplished through the low pressure turbine shaft. The shaft design requirements are shown in Table III. The shaft configuration, shown in Figure 24, is machined from an Inco 718 forging. The shaft is designed for 65,531 N-m (580,000 in.-lb) of torque for a minimum life of 10,000 takeoff cycles. A temperature distribution along the shaft (Figure 25) was determined to establish the material allowables at the critical sections of the shaft. The temperatures shown in Figure 26 are those expected 270 seconds after a throttle movement from idle to takeoff conditions.

Figures 26 and 27 show the results of the stress analysis of the shaft. All stresses are effective tensile stresses which are compared to the tensile capability of Inco 718. Both high cycle fatigue (HCF) and low cycle fatigue (LCF) have been evaluated. The limiting criteria is HCF in the shaft portion with the LCF life being in excess of the 10,000 cycle life requirement. The results of the stress analysis of the aft cone area are shown in Figure 27. LCF life at Points 5 and 10 is the limiting criteria. Creep and rupture life are greater than 10,000 hours and 100,000 hours, respectively, and do not limit the shaft design.

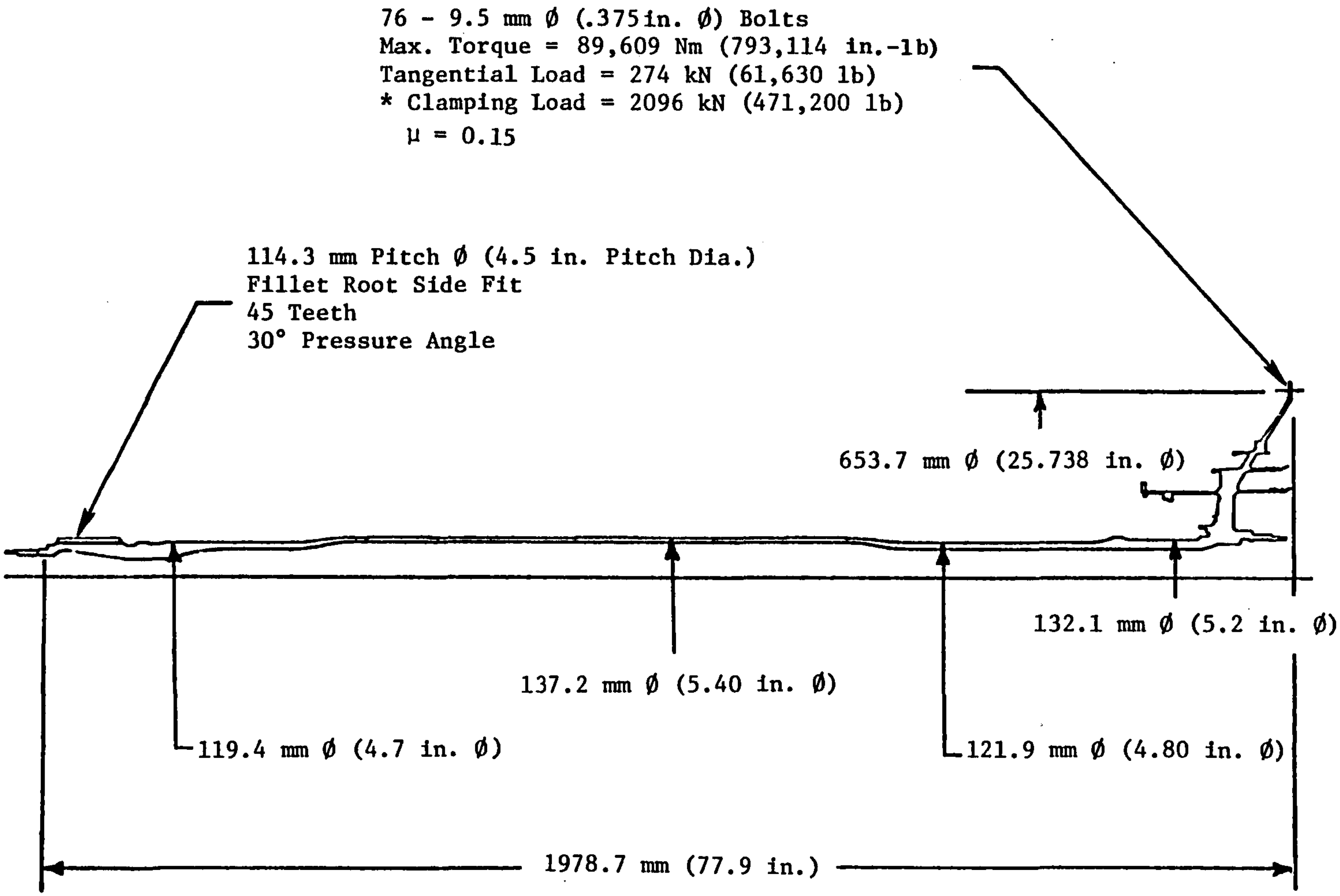
The forward spline interface with the forward fan shaft is shown in Figure 28. The spline has been configured utilizing the tooth design from another engine program; this allows the possibility of utilizing existing tooling. The spline life is limited in LCF and has been calculated at 16,000 cycles. The inside bore and the outside diameter of the LPT shaft and forward shaft have been configured to facilitate torque transfer from one shaft to the other.

Table III. Low Pressure Turbine Shaft Design Requirements.

	ICLS	FPS
Required Life	10,000 Takeoff Cycles	36,000 Flight Cycles
Maximum Torque Nominal	65,407 N-m (578,900 in.-lb)	77,925 N-m (689,700 in.-lb)
Design Life	10,000 Takeoff Cycles*	50,000 Flight Cycles**
Design Loads	1 "g" Bending, Misalignment (Bearings)	1 "g" Bending, Misalignment (Bearings)
	Turbine Thrust Loads, Steady-State and Alternating Torques	Turbine Thrust Loads, Steady-State and Alternating Torques
		Flight Manuver Bending Loads

* With 1 Peak Stress Cycle Per Test Cycle

** With 2 Peak Stress Cycle Per Flight



* Relaxed Load After 9000 hours
1827.5 kN (410,831 lb) Required

Figure 24. Low Pressure Turbine Shaft Configuration.

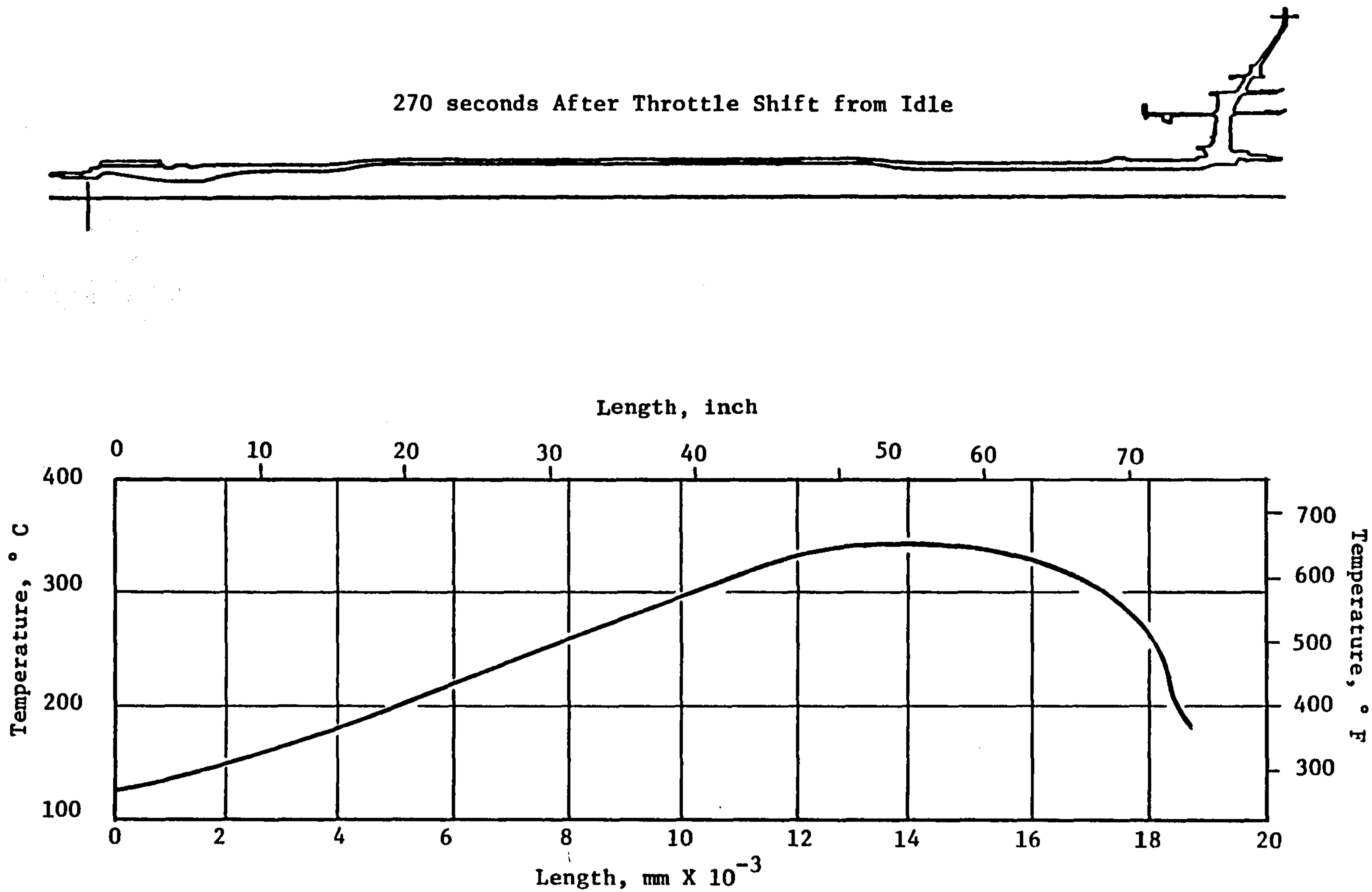
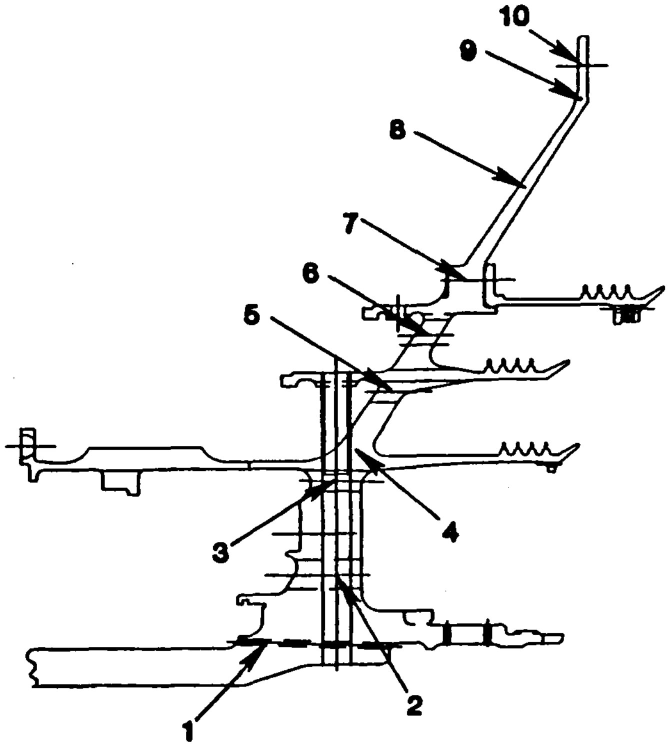


Figure 25. Low Pressure Turbine Shaft Temperature at Takeoff.

ICLS-(I-718)

Location	1	2	3
Mean Stress (HCF) Effective Tensile	551.6 MPa (80 ksi)	620.5 MPa (90 ksi)	620.5 MPa (90 ksi)
Alt. Stress Effective Tensile	186.2 MPa (27 ksi)	158.6 MPa (23 ksi)	172.4 MPa (25 ksi)
Alt. Stress Allowable -10^7 Cycles	186.2 MPa (27 ksi)	172.4 MPa (25 ksi)	172.4 MPa (25 ksi)
Low Cycle Fatigue Life (Cycles)	4.5×10^4	2×10^4	2.5×10^4
Mean Stress (LCF) Effective Tensile A = 1	448.2 MPa (65 ksi)	413.7 MPa (60 ksi)	434.4 MPa (63 ksi)
Temperature	149° C (300° F)	343° C (650° F)	315° C (600° F)

Figure 26. Low Pressure Turbine Shaft Stresses.



Based on 60 seconds Accel to T/O From Idle.

Material-Inconel 718

Point	Temp. ° C (°·F)	LCF Stress for A = 1 MPa (ksi)	LCF, Cycles	σ Eff. (max) No Kt, MPa (ksi)	0.2% Creep Life (hour)	Rupture Life (hour)
1	102.2 (216)	444.7 (64.5)	15,000	296.5 (43.0)	-	-
2	101.1 (214)	447.7 (64.5)		296.5 (43.0)	-	-
3	119.4 (247)	434.4 (63.0)		496.4 (72.0)	-	-
4	182.2 (360)	337.8 (49.0)	>100,000	675.6 (98.0)	-	-
5	371.1 (700)	465.4 (67.5)	10,000	310.3 (45.0)	-	-
6	425.6 (798)	475.7 (69.0)		317.1 (46.0)	-	-
7	502.8 (937)	258.5 (37.5)	>100,000	172.4 (25.0)	-	-
8	536.6 (997)	86.2 (12.5)		172.4 (25.0)	>10,000	>100,000
9	555 (1031)	382.6 (55.5)	50,000	510.2 (74.0)		
10	576.1 (1069)	455.0 (66.0)	10,000	303.4 (44.0)		

Figure 27. Low Pressure Turbine Shaft Aft Cone Stresses.

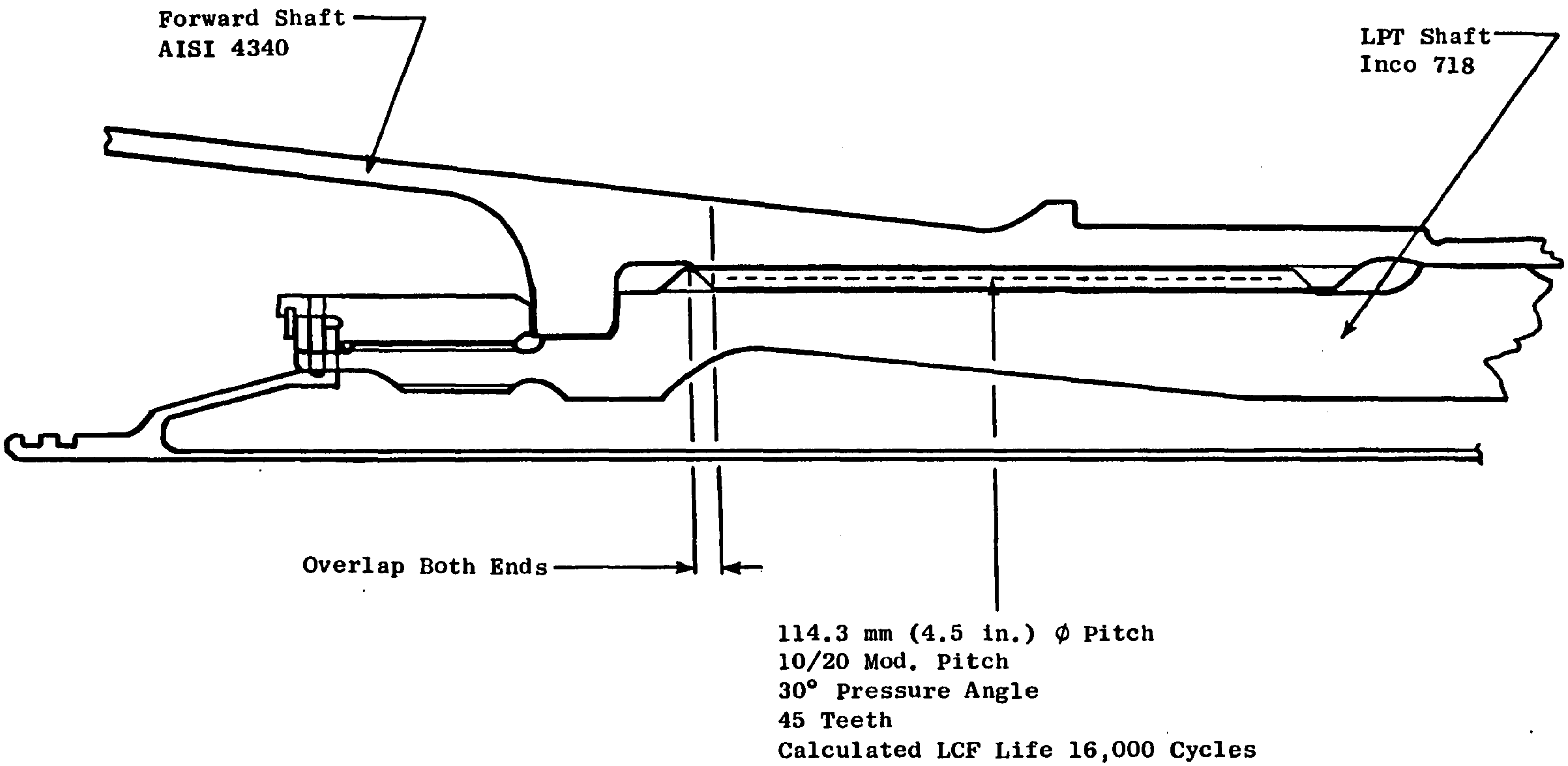


Figure 28. Low Pressure Turbine Shaft Forward Spline Design.

6.0 DRIVE SYSTEM

The drive system schematic is shown in Figure 29. The PTO gearbox mounted in the forward sump drives the accessory gearbox (AGB) mounted on the engine outer casing. The AGB provides drive pads for the following accessories:

- Lube and Scavenge Pump
- Two Air Starters
- Control Alternator
- Fuel Pump/Control

The gearbox is designed for a maximum combined torque from the starters of 1084.6 N-m (800 lb-ft). The expected maximum accessory hp requirements for the ICLS engine is 53.7 kW (72 hp). This is much lower than a typical commercial engine which may be as high as 335.6 kW (450 hp) due to aircraft required generators. The drive system design for the ICLS engine is identical to that of the core engine with the exception of the radial drive shafting.

Below is a comparison of the design requirements of the FPS and the core/ICLS engine.

Table IV. Design System Design Life Requirements.

	Core/ICLS Engine	FPS Engine
Total No. of Starts	2000	40,000
Total Service Life	2000 hr	36,000 hr

6.1 POWER TAKEOFF DESIGN

The PTO is shown in Figure 30. The PTO gear is supported by three bearings. Shown in this figure is the penetration required to assemble the PTO through the support housing. The configuration of the slot in the support housing is sized to provide for assembly. Two roller bearings react the tangential loads and the loads caused by the overturning moments. A thrust bearing reacts the gear thrust. The outer rings, inner rings, and rolling elements are M-50 material; the cages are silver-plated AMS 6414 (AISI 4340). The bearings for the PTO are sized to expected FPS loads, and the calculated lives shown in Figure 31 meet the design goals of 36,000 hours.

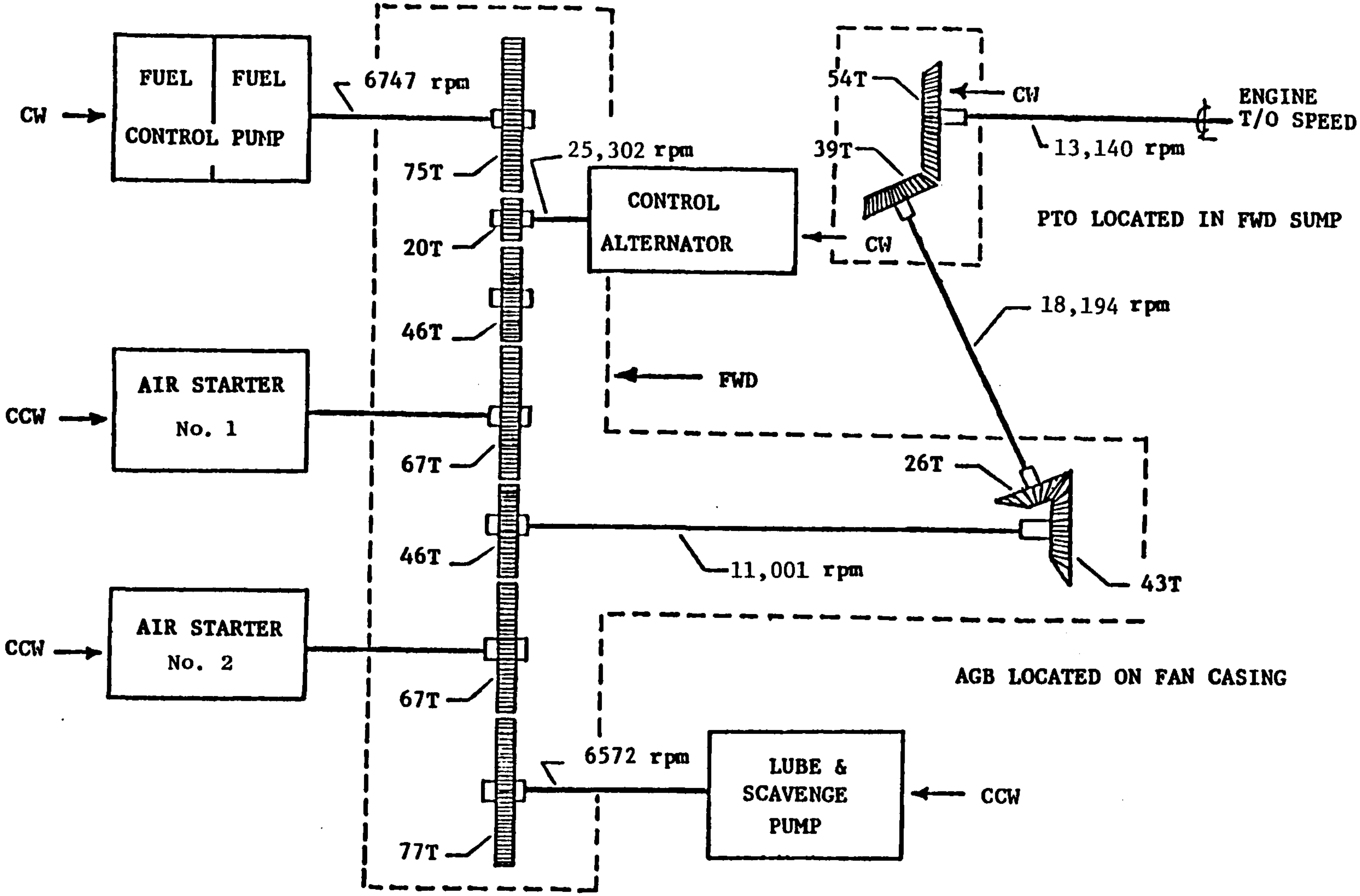


Figure 29. ICLS Accessory Drive Schematic.

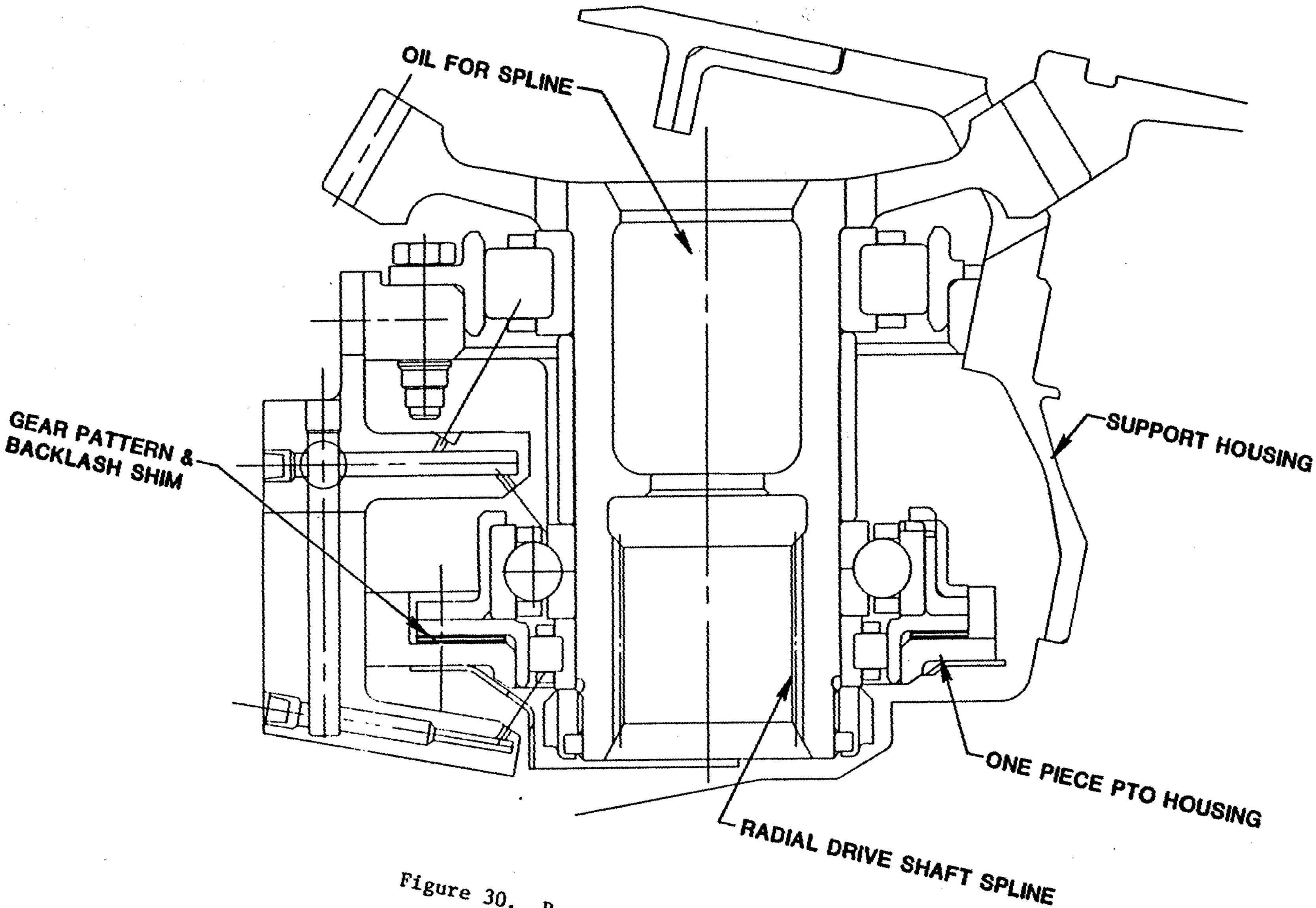
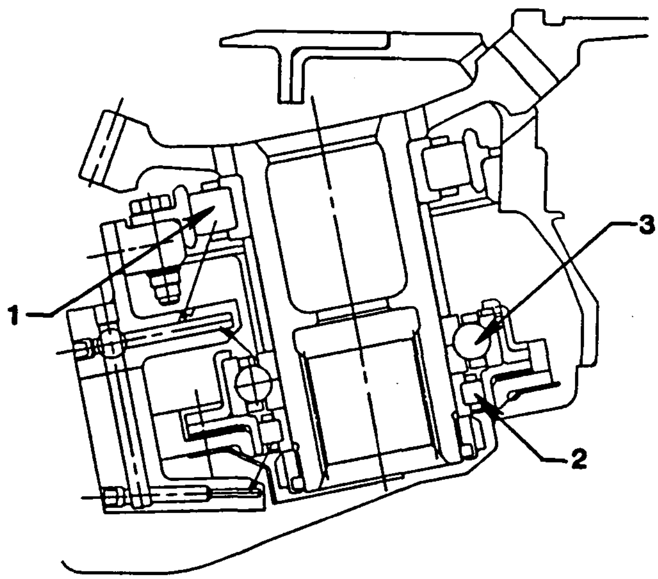


Figure 30. Power Takeoff Gearbox.



12,282 Rotor rpm

	POSITION		
	1	2	3
Size	211	1911	111
Average Speed, rpm	17,007	17,007	17,007
FPS Loads,	2958 N (665 lbs)	1646 N (370 lbs)	1664 N (374 lbs)
Life, hour	127,200	97,376	76,419

System Life - 45,970 Hours

FPS Design Requirements - 36,000 Hours

Figure 31. PTO Bearing Design Summary.

A summary of the bevel gear design is shown in Figure 32. These gears are also designed for FPS starting and accessory load conditions. The bending and compressive stresses are below the allowable stresses for the material. The shaft angle of 78° 03' was chosen to facilitate mounting the FPS accessory gearbox in the core compartment area.

Although the gear pitch line speed is 7% higher than current practice, it should not be a problem. Special attention is being given to the location of the gear mesh lubrication and cooling jets. Two are being used in proximity to the gear mesh on both the incoming and outgoing side of the mesh.

The bevel gear contact patterns and backlash are being developed and specified to allow for slight radial movements of the horizontal gear due to spring-mounted rotor system.

6.2 ACCESSORY GEARBOX DESIGN

The AGB design is shown in Figures 33 and 34. The AGB is being designed to meet the requirements of both the core and ICLS engine; the only difference is in the mounting of the gearbox to the engine casing.

The gearbox features a one-piece main housing with four adapters to which gears and bearings are preassembled before final assembly into the gearbox housing. Three of the seven spur gears have been obtained from other engine programs and three gears have been designed specifically for the Energy Efficient Engine gearbox. Only four new bearings will be required for the gearbox and many of the miscellaneous smaller parts have also been obtained from other programs.

The material for the housing and adapters will be 356-T6 aluminum, the gears will be AISI 9310, and the bearings are made of M-50 with silver-plated AISI 4340 cages.

A summary of the bearing loads, speeds and lives for both the spur and bevel gears is shown in Figures 35 and 36. The lives of the gearbox bearings are more than adequate for the core and ICLS engine with the minimum bearing life being 400,000 hours.

The spur and bevel gear design information is shown in Figure 37 and Table V. The bevel gears are sized for 2000 starts assuming that each start utilizes the maximum torque of the starters, which is conservative. The bending strength of the bevel gears is enhanced by utilizing a 22.5° pressure angle and a 35° spiral angle which gives a relatively high face contact ratio.

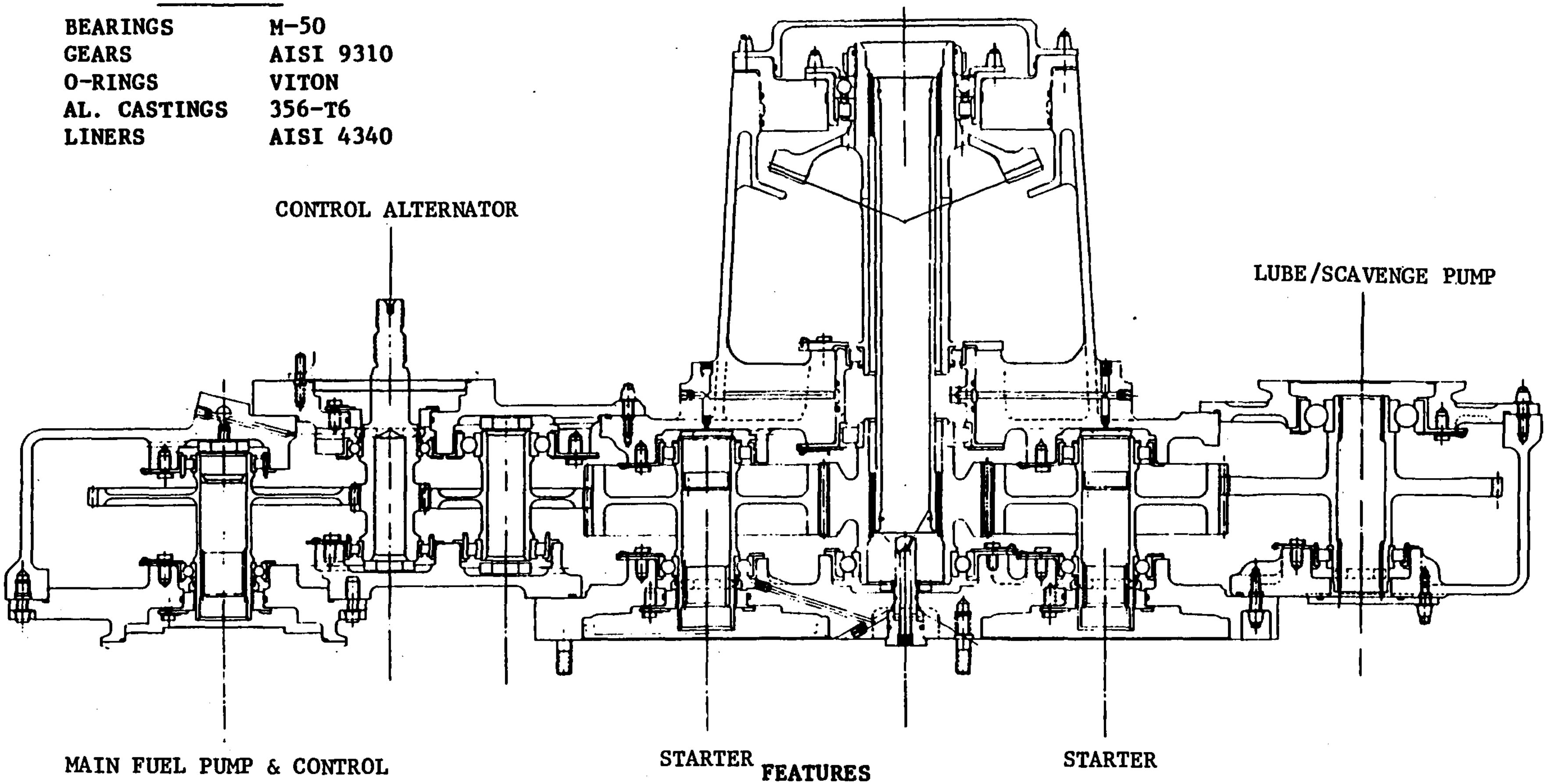
The new spur gear diametral pitch must be 10 to mesh with the existing gears. The face width of the starter gears was determined from the starting requirements; 2000 starts are attainable without exceeding the allowable bending stress. The face width of the starter gears is 50.8 mm (2.0 in.) and the 46T input gear is provided with crowning of 0.01225 to 0.02940 mm (0.0005 to 0.0012 in.) to prevent edge loading of teeth.

• Gear Tooth Number	
- Pinion	39
- Gear	54
• Face Width	23.4 mm (0.92 in.)
• Diametral Pitch	6.488
• Pressure Angle	22.5°
• Spiral Angle	35.0°
• Shaft Angle	78° 03'
• Max. Surface Speed	8839.2 m/min (29,000 fpm)
• Bending Stress	
- Start	255.1 MPa (37.0 ksi) [258.5 MPa (37.5 ksi) Allowable]
- Max. Accessory Load (FPS Load)	68.9 MPa (10.0 ksi) [258.5 MPa (37.5 ksi) Allowable]
• Compressive Stress	
- Start	1365.1 MPa (198.0 ksi) [1723.6 MPa (250.0 ksi) Allowable]
- Max. Accessory Load (FPS Load)	690.8 MPa (100.2 ksi) [1723.6 MPa (250.0 ksi) Allowable]
• Scoring	ΔT is Low (No Problem)

Figure 32. PTO Bevel Gear Design Summary.

MATERIALS

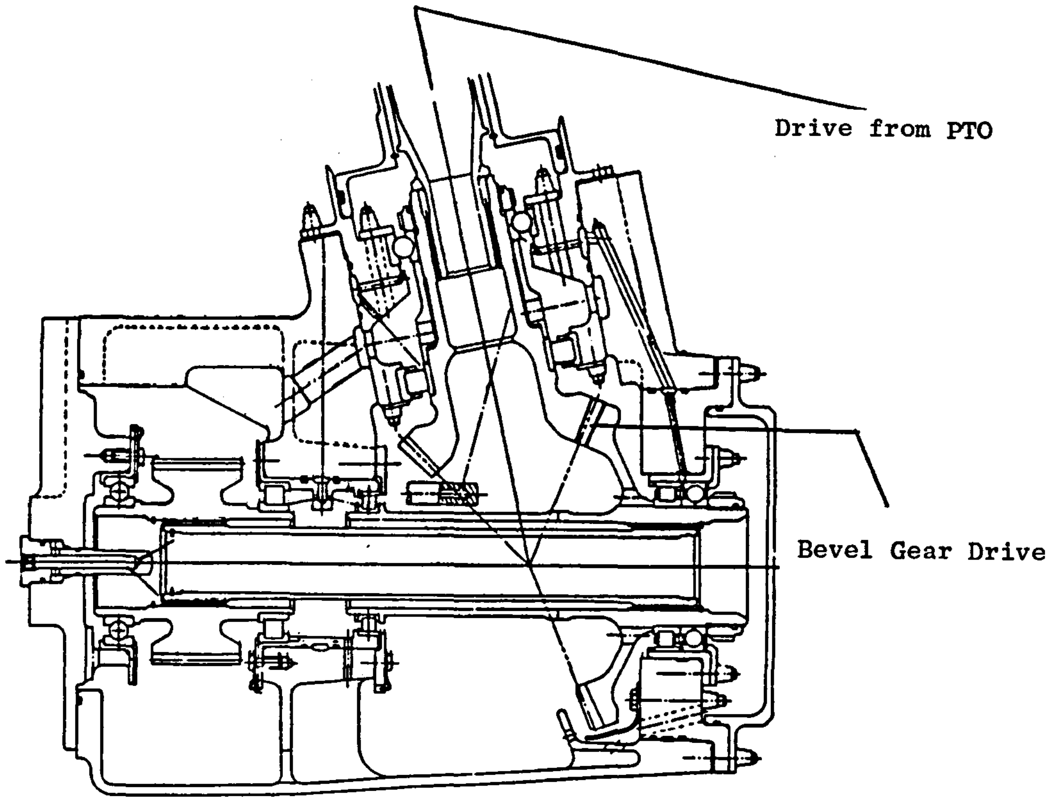
BEARINGS	M-50
GEARS	AISI 9310
O-RINGS	VITON
AL. CASTINGS	356-T6
LINERS	AISI 4340



FEATURES

- ONE-PIECE MAIN CASING
- HARDWARE COMMON WITH OTHER GE ENGINES
- CARBON SEALS - MATING RINGS OIL COOLED
- THREE NEW SIMPLE ADAPTORS

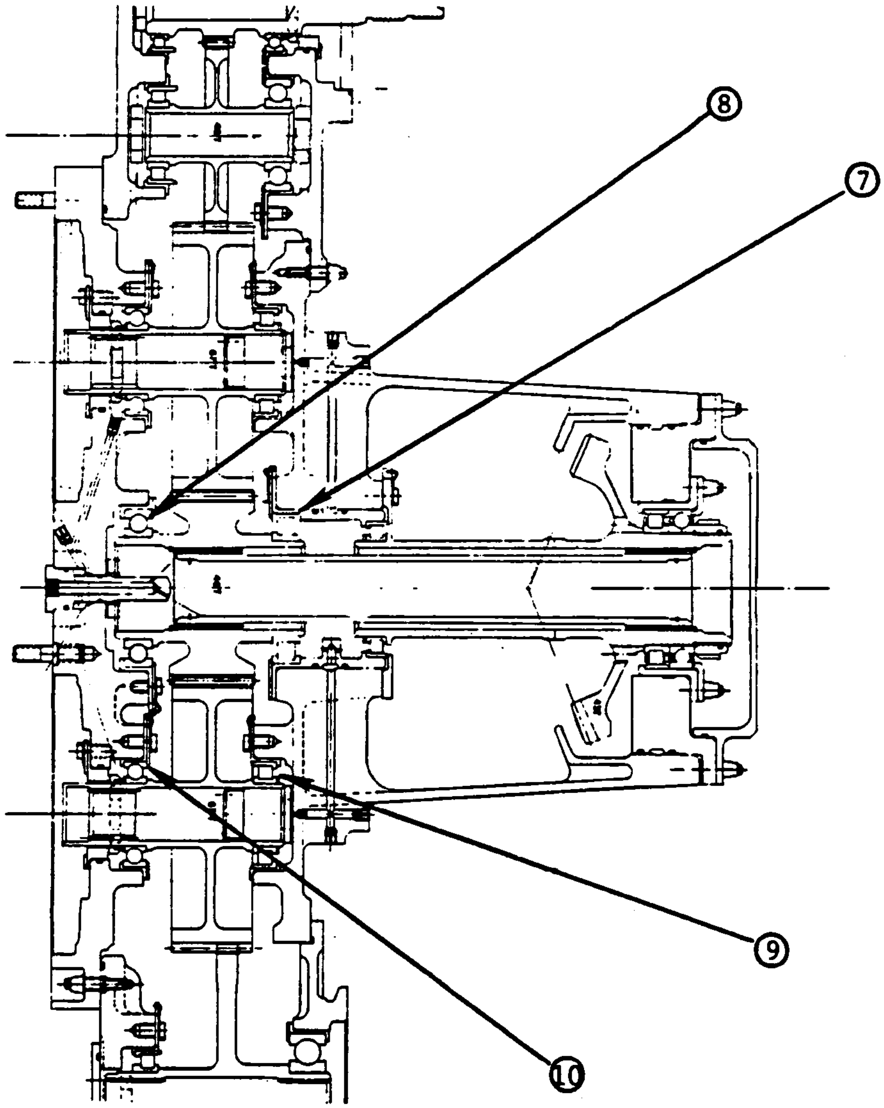
Figure 33. Core/ICLS Accessory Gearbox.



<u>Features</u>

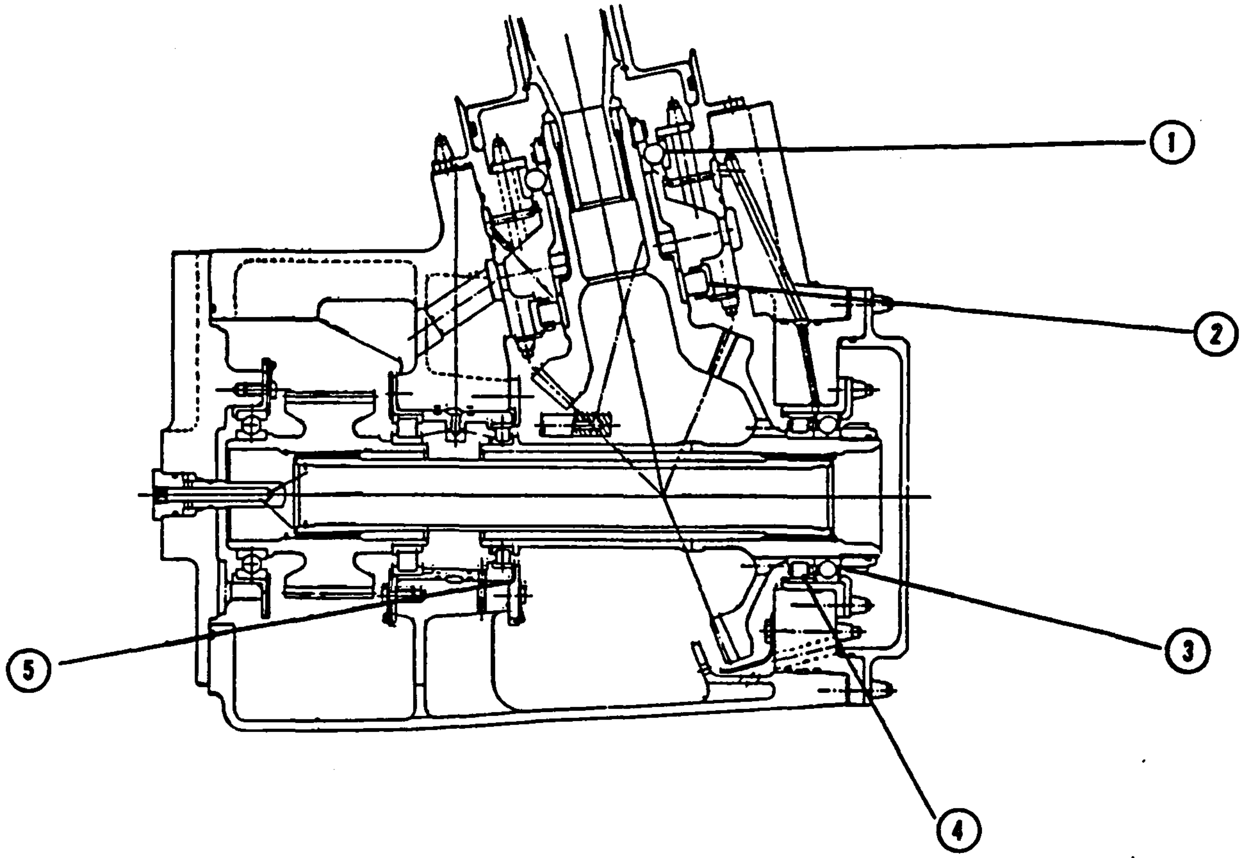
- Bevel Gear Jet Lubed In and Out of Mesh
- Bearing Jet Lubricated
- All Splines Lubricated

Figure 34. Accessory Gearbox Bevel Gear Cross Section.



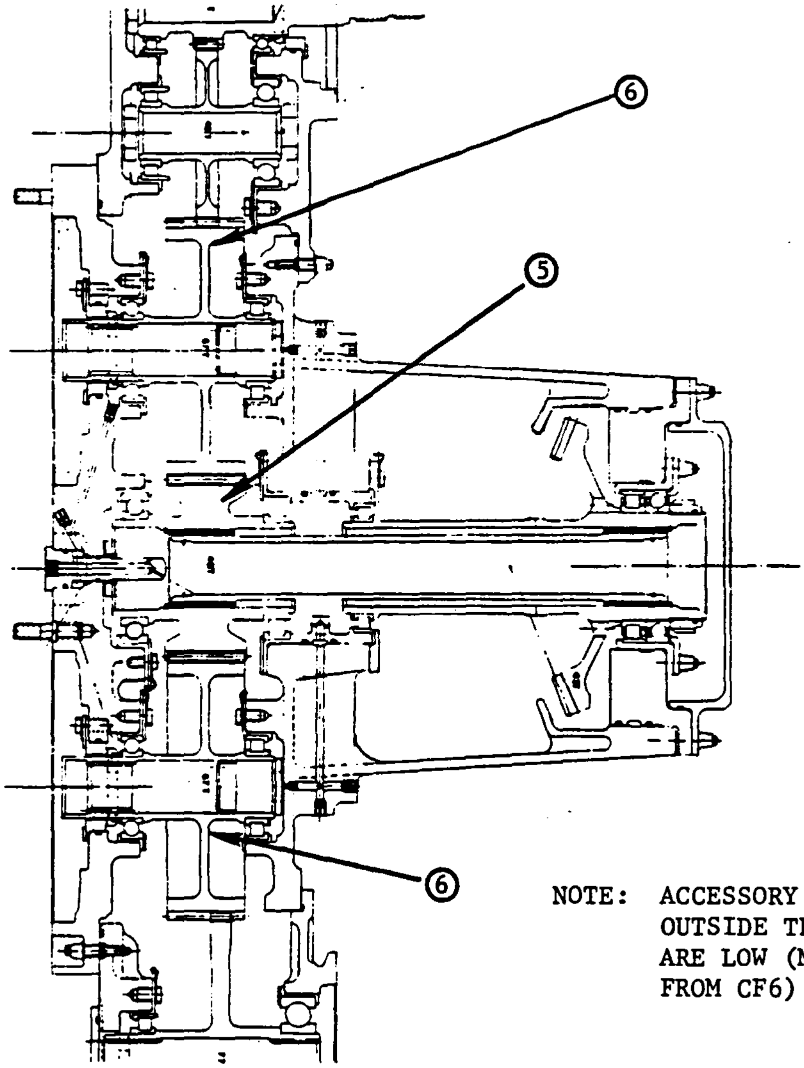
POSITION	7	8	9	10
T/O rpm	11,000	11,000	7,550	7,550
CML, N (lbs)	423 (95)	423 (95)	391 (88)	200 (45)
Life, hr	43×10^6	0.6×10^6	1.9×10^6	0.4×10^6

Figure 35. Core/ICLS AGB Spur Gear Bearing Design Summary.



POSITION	1	2	3	4	5
SIZE	111	114	014	014	012
T/O rpm	18,194	18,194	11,000	11,000	11,000
T/O Radially/Axially N (lbs)	676/-0.227 (152/-51)	1655 (372)	0/560 (0/126)	552 (124)	414 (93)
B_{10} Life (hr.)	0.28×10^6	0.28×10^6	2.1×10^6	9.0×10^6	3.8×10^6

Figure 36. Core/ICLS AGB Bevel Gear Bearing Design Summary.



NOTE: ACCESSORY SPUR GEAR STRESSES OUTSIDE THE STARTER DRIVE TRAIN ARE LOW (MOSTLY EXISTING HARDWARE FROM CF6)

• Gear Tooth Numbers	
5	46
6	67
• Face Width	50.8 mm (2.0 in.)
• Diametral Pitch	10
• Pressure Angle	20°
• Bending Stress	
Start (2000 Starts)	492.2 MPa (71,400 psi)
Max. Accessory Load	76.5 MPa (11,100 psi)
• COMPRESSIVE STRESS	
Start	1309.9 MPa (190,000 psi)
Max. Accessory Load	515.7 MPa (74,800 psi)
• Scoring	ΔT Is Low

Figure 37. Core/ICLS AGB Spur Gear Design Summary.

Table V. Core/ICLS AGB Bevel Gear Design Summary.

• Gear Tooth Numbers	
– Pinion	26
– Gear	43
• Face Width	(1.10 in.) 27.9 mm
• Diametral Pitch	5.60
• Pressure Angle	22.50°
• Spiral Angle	35.0°
• Shaft Angle	101° 57'
• Max. Surface Speed	(22,100 fpm) 6736.1 m/min
• Bending Stress	
– Start (2000 Starts)	(38,270 psi) 263.8 MPa
– Max. Accessory Load (FPS loads)	(8450 psi 37,000 psi Allowable)
	58.3 MPa 258.5 MPa Allowable
• Compressive Stress	
– Start	(227,300 psi 250,000 psi Allowable)
	1567 MPa 1723 MPa Allowable
– Max. Accessory Load (FPS loads)	(106,800 psi 250,000 psi Allowable)
	736.3 MPa 1723 MPa Allowable
• Scoring	ΔT is Low (No Problem)

6.3 RADIAL DRIVE SHAFT

Detail information concerning the radial drive shafting and the midspan bearing assembly is shown in Figures 38 and 39, respectively. The longer, lower shaft diameter is determined from the shaft first critical frequency, which is kept at least 20% above the maximum operating speed of the shaft. The shorter upper shaft diameter is determined from the torque to be transmitted during starting conditions.

A midspan bearing assembly provides support for the two shafts between the core and fan flowpath. A roller bearing is used to facilitate ease of assembly. The bearing is provided with its own lube jet on the lower side of the bearing to enhance oil drainage from the bearing area. It is expected that only small amounts of oil will drain down from the midsump and a flowpath is provided around the bearing to handle this flow.

The shafting assembly is provided with an arrangement to prevent the shaft from dropping out while accessory gearbox assembly is being mounted.

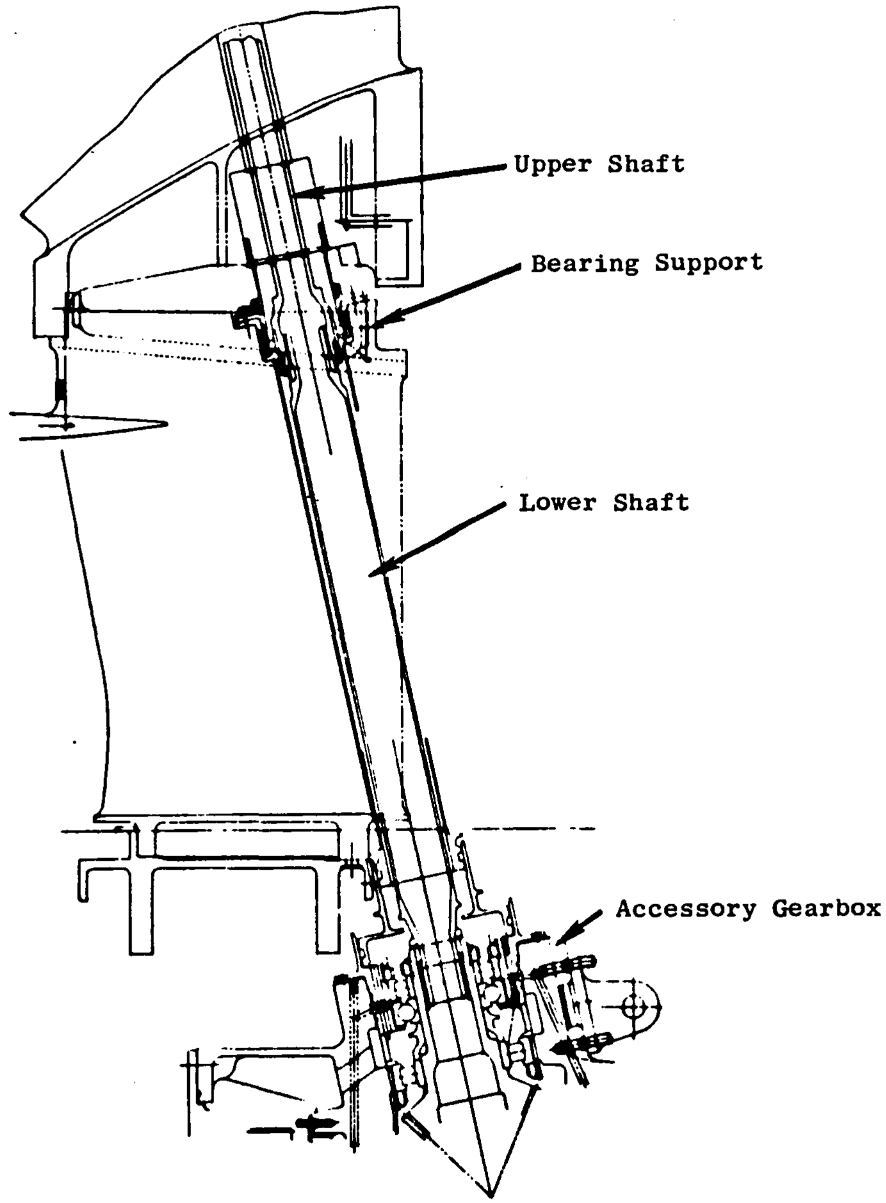


Figure 38. Accessory Drive System Radial Drive Shaft.

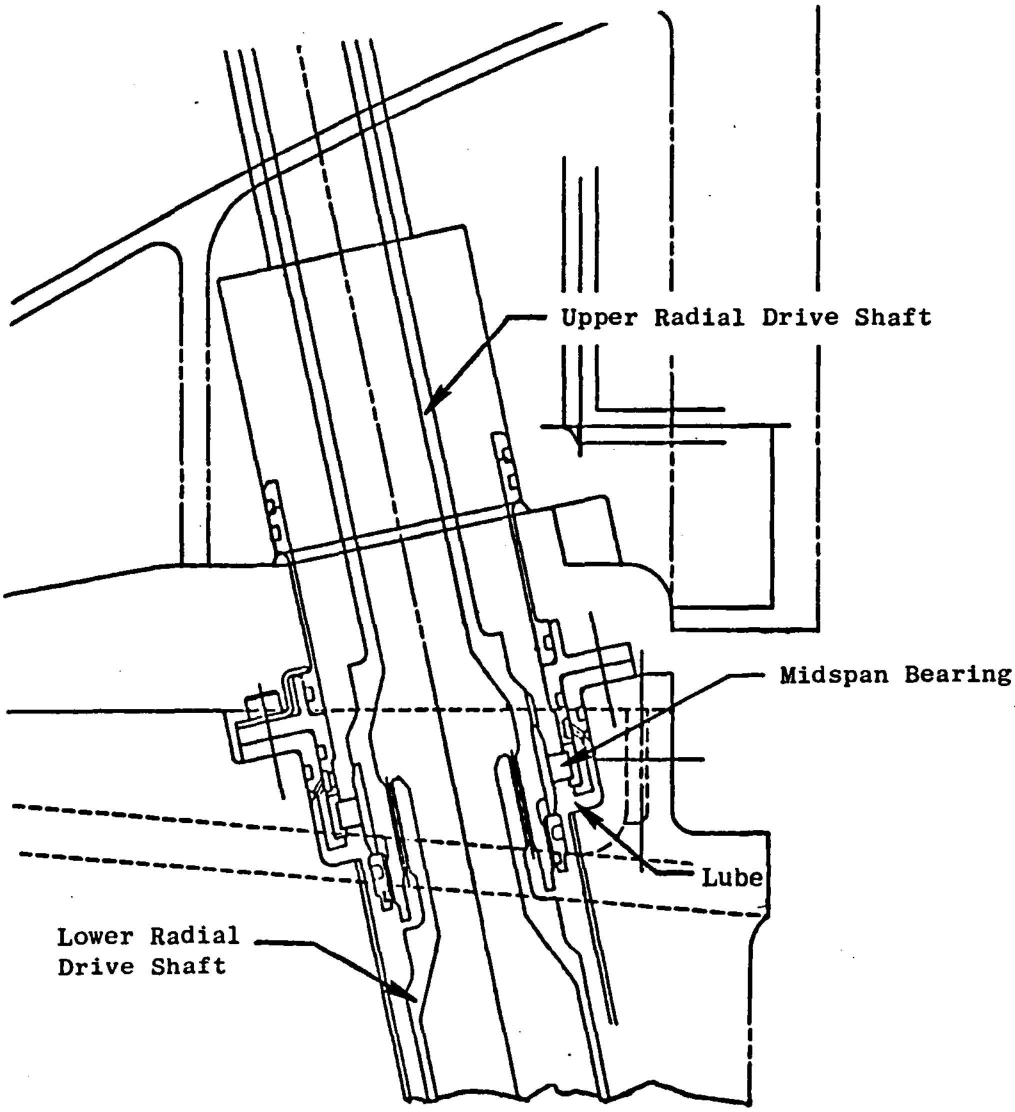


Figure 39. Accessory Drive System Midspan Bearing Support.

7.0 LUBE SYSTEM

The lube system is described in the schematic shown in Figure 40 and is typical of systems used in other General Electric engines.

The engine lube and scavenge pump mounted on the accessory gearbox supplies oil to each sump, the gearbox and the radial drive shaft bearing. The forward and aft sump and the gearbox are scavenged by individual pumping elements also provided in the pump. The flow distribution to each area and the scavenge capacity of the scavenge elements are shown in Table VI.

Table VI. Lube Supply Versus Scavenge Pump Capacity.

	Lube Specified	Scavenge Capacity
Fwd. Sump	23.4 lpm (6.2 gpm)	65.8 lpm (17.4 gpm)
Aft Sump	9.8 lpm (2.6 gpm)	29.3 lpm (7.75 gpm)
Accessory Gearbox	11.0 lpm (2.9 gpm)	42.4 lpm (11.2 gpm)
Bypass Oil	19.1 lpm (5.05 gpm)	
	63.3 lpm (16.75 gpm)	

The pump being used for the ICLS engine is an existing pump from another engine program. The excess capability of this pump is the oil bypassed as shown in the above table and Figure 40. The bypassed oil is directed back to the inlet of the pump.

There are two scavenge pumping elements that will not be used and will be blanked off during testing. Oil from the common scavenge manifold will "wet" these unused elements during operation.

Oil filters are used on both the supply and scavenge side of the lube system. The supply filter protects the sumps and gearbox from contamination and the scavenge filters protect the heat exchanger and the lube tank. Each scavenge element also has an inlet screen to protect the pumping elements from larger debris.

The lube tank and the heat exchanger are also from other engine programs. Excess air from the scavenge system is returned to the engine by the tank vent line; a check value is used to maintain the tank pressure 68.9 KPa (10 psi) above the sump pressure. Check valves are used to prevent draining the tank into the sumps causing flooding at shutdown.

The engine sumps are center vented to the exhaust of the engine through air/oil separators located in the forward and aft sump.

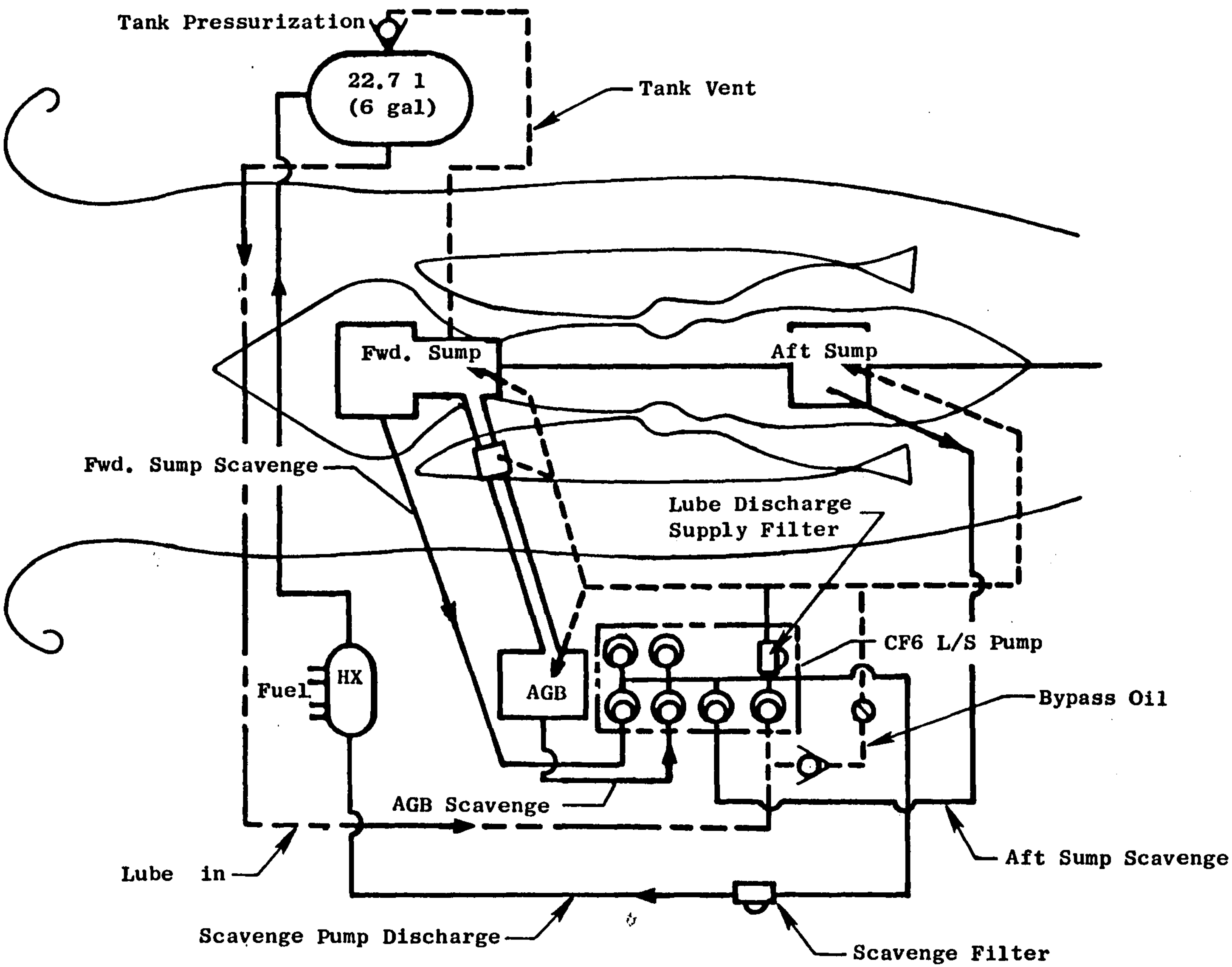


Figure 40. Lube System Schematic.

8.0 SECONDARY AIRFLOWS, PRESSURES AND ROTOR THRUST SYSTEM

The sump labyrinth seals for the ICLS engine are pressurized by air from the core inlet flowpath, and air passages are fed from the leading edge of the core forward frame struts. Air in these passages is directed into a plenum which supplies pressurization to the forward seals after which air is directed to the aft sump through an annulus formed by the OD of the center vent lube and the ID of the low pressure turbine (LPT) shaft. The labyrinth seal just aft of the No. 3 bearing is pressurized from the hub of the core flowpath just before the air enters the compressor. Figures 41 and 42 show pressures and flows in various areas of the engine pertinent to the sumps at maximum test cell conditions. The minimum ΔP across the forward and aft sump seals are 22.06 KPa (3.2 psid) and 13.79 KPa (2.09 psid), respectively, which is in line with current practice. At idle conditions, seal ΔP should be greater than 101.6 mm (4.0 in.) of water which is a minimum General Electric limit.

An analysis has been performed to determine the characteristics of the thrust load on the No. 1 (HP system) and No. 3 (HP system) thrust bearings. The results of this analysis are shown in Figure 43. The thrust load on the No. 1 bearing will vary between +4097 to -53,379 N (+921 to -12,000 lb) from idle to T.O speed (plus indicates a forward load). The No. 3 bearing load varies from +912 to +26,689 N (+205 to +6000 lb) between idle and T/O conditions.

Pressures and Airflows

Max. Test Cell Conditions

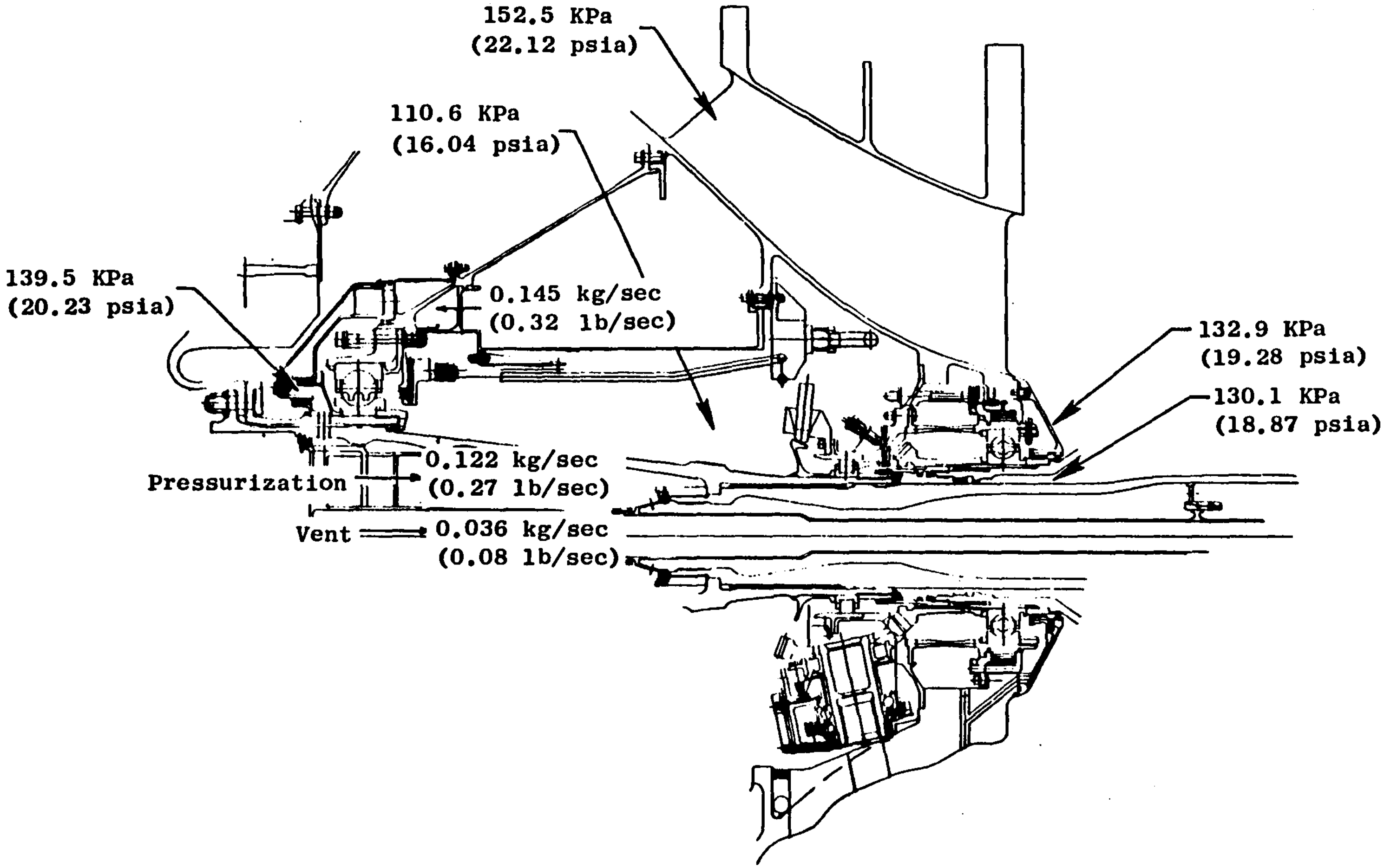


Figure 41. Secondary Air System, Forward Sump.

Pressures and Airflows
Max. Test Cell Conditions

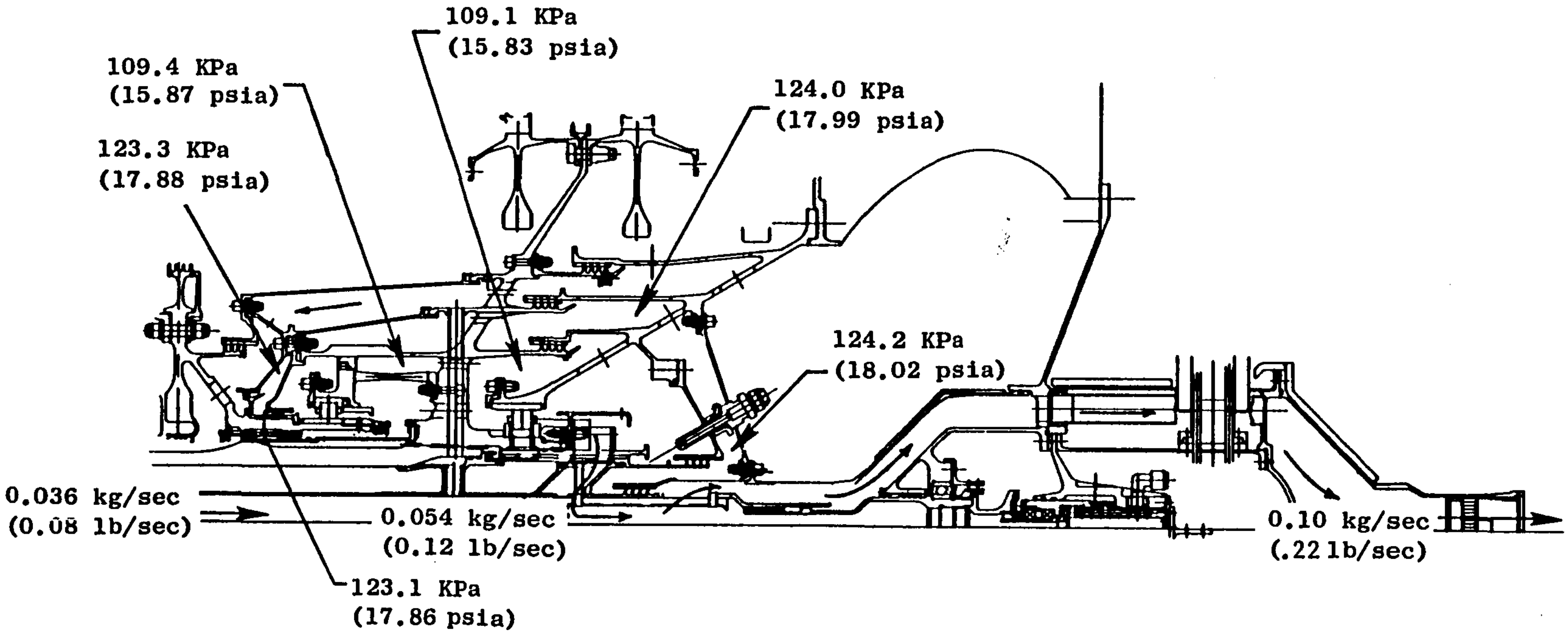


Figure 42. Secondary Air System, Aft Sump.

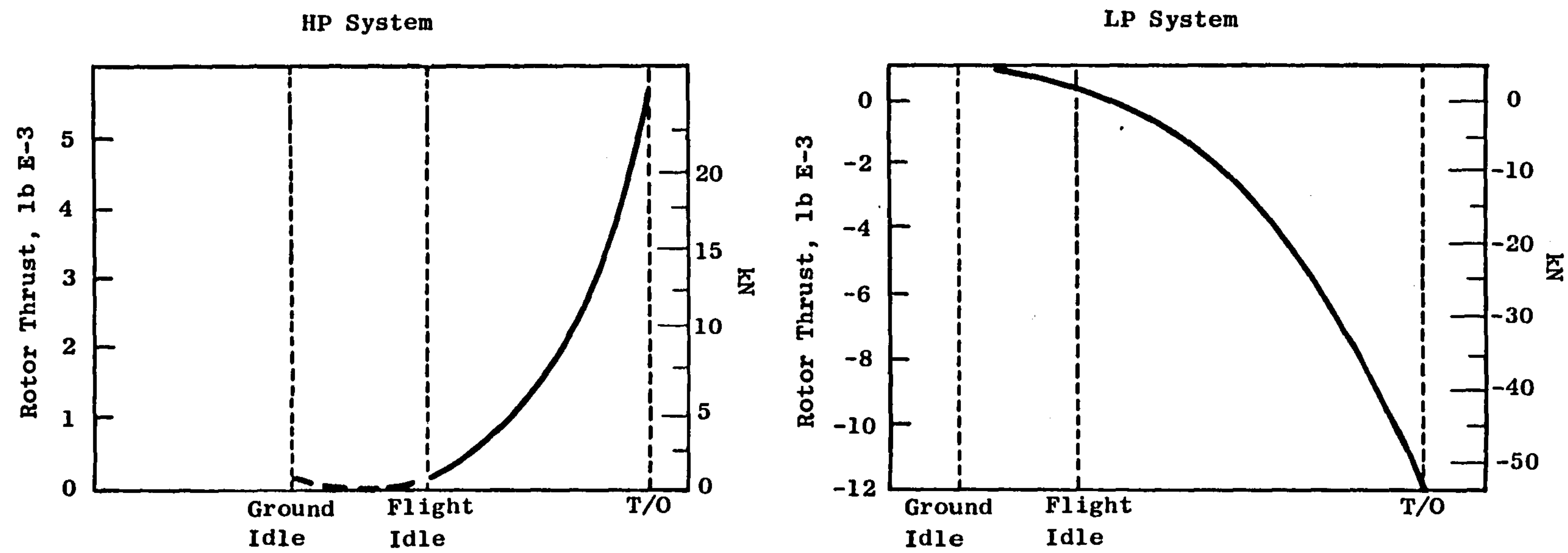


Figure 43. ICLS Engine Rotor Thrust.

9.0 CONFIGURATION DESIGN

The configuration design effort encompasses the following areas:

1. The large pneumatic piping design required for the compressor and turbine clearance control.

2. Design of the starting air manifolds and determining the location of the starting bleed valves.

3. Design of all the lube and fuel lines.

4. The electrical harness design.

5. The design of the isolation box needed to provide service line strut penetration.

6. The design of the core compartment partition wall.

The first two items represent the major design effort because of their large physical size and the restrictions on their location. This piping is shown schematically in Figure 44 and much of this hardware was originally designed for the core engine. The major piping configured on the engine is shown in Figure 45.

The air from the fifth stage of the compressor is utilized for compressor clearance control. This air can be directed internally over the compressor aft inner casing or externally through piping. Whether the air is flowing internally over the inner compressor casing or externally through piping is determined by the compressor clearance control bypass valve which is actuated by the engine control system.

Air from the compressor clearance control system is directed to the low pressure turbine purge cavity manifold where it is used for cooling. Check valves are provided to prevent any possibility of air backflow. A fifth stage customer bleed was originally provided but has been eliminated due to redefinition of requirements.

A manifold is also provided for the seventh stage air and porting from the manifold serves two functions. Compressor flowpath air bleed is provided during starting and cooling air is directed by piping to the second stage vane of the high pressure turbine rotor. This flow circuit is also protected against air backflow by check valves located in the four pipes leading to the cooling manifold for the high pressure turbine second stage vanes.

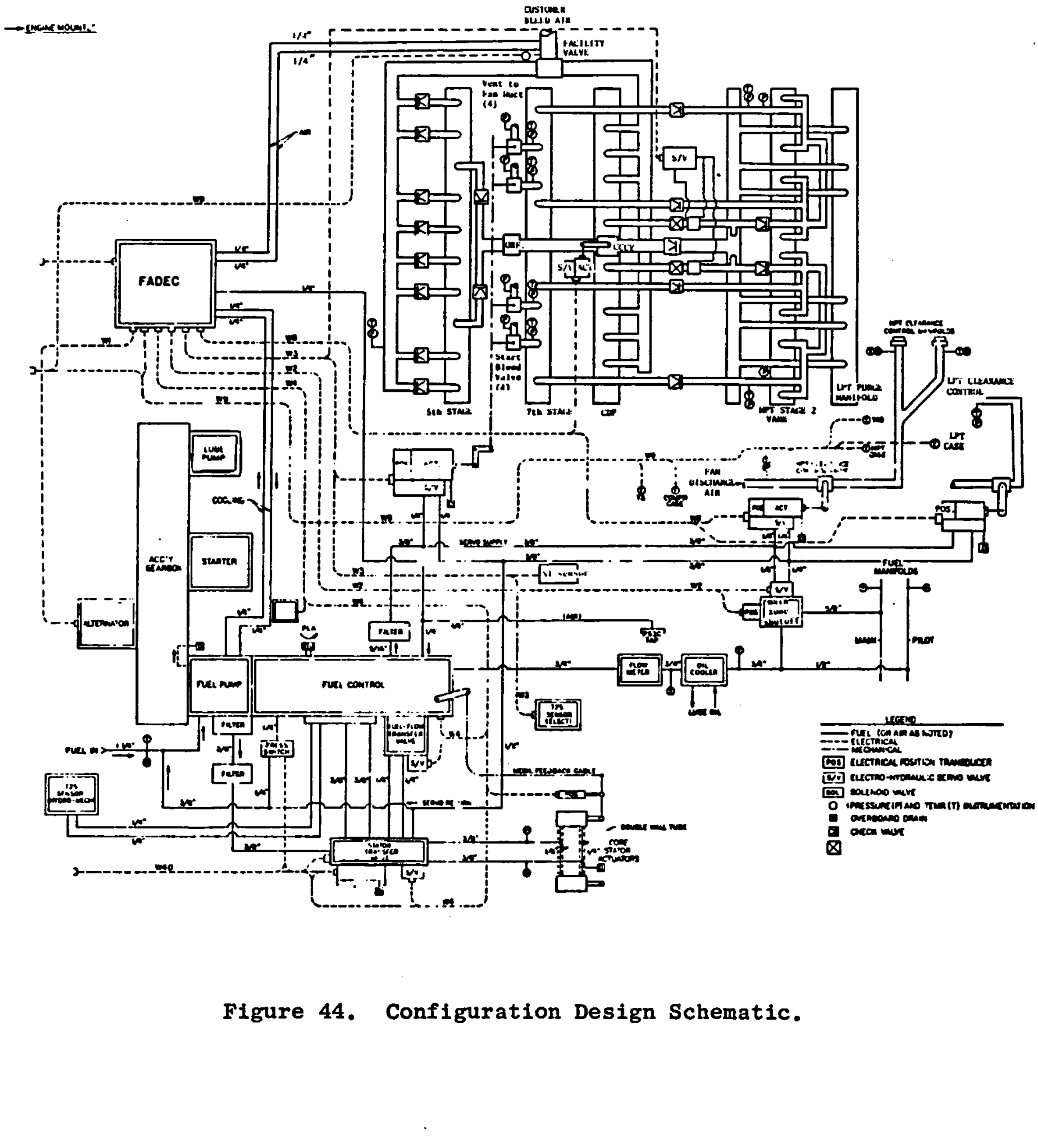


Figure 44. Configuration Design Schematic.

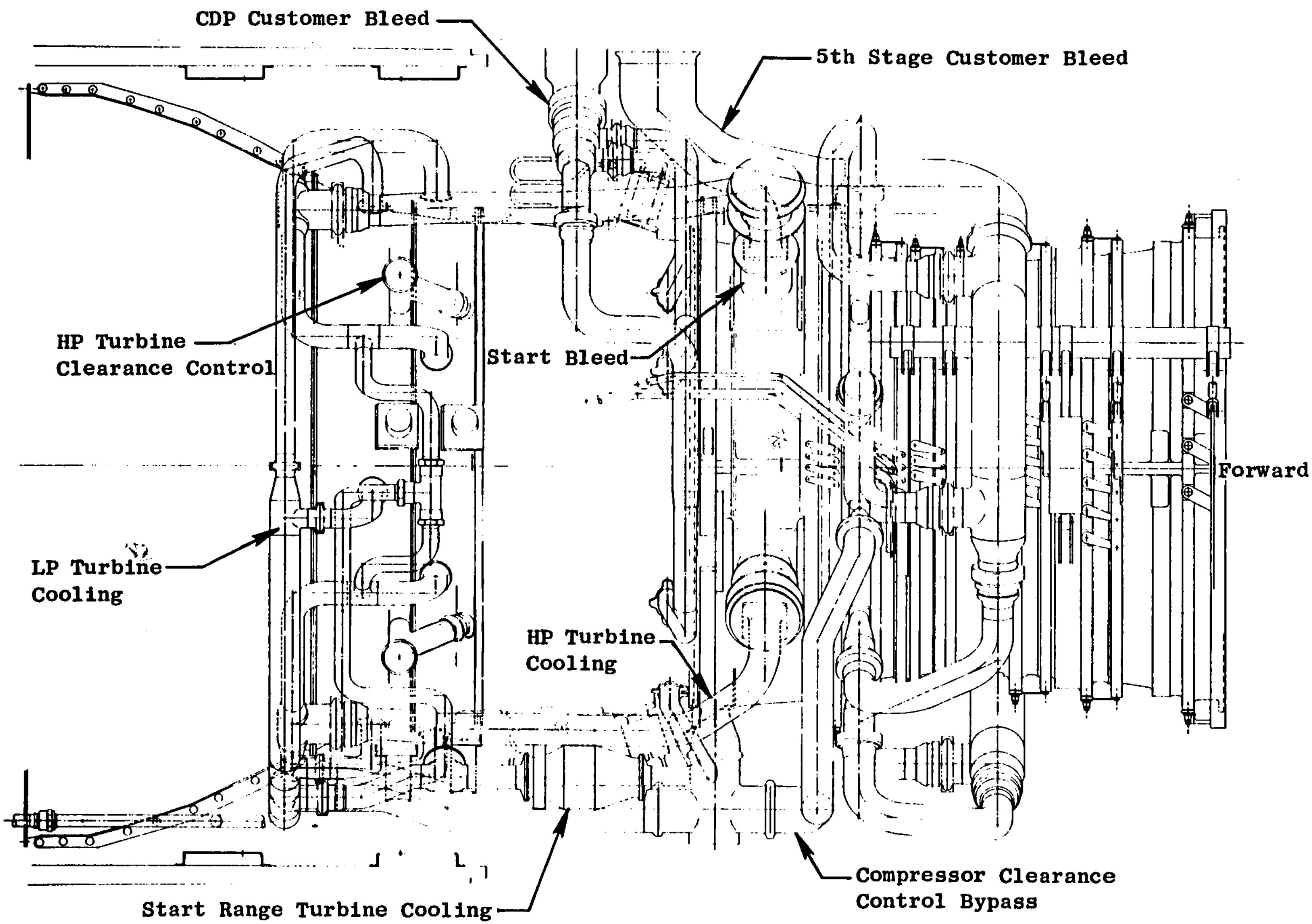


Figure 45. Configuration Design.

The design analysis of these large pipes used the same design techniques that have been successfully used on other General Electric engines. Careful attention has been paid to a thermal analysis of the piping to help locate clamps to prevent excessive vibration and yet not constrain the pipes which will induce severe stresses.

It is planned to determine the vibration characteristics of the major piping when assembled on the engine. This vibration test will establish if additional restraints are needed for the piping.

The design approach used for the lube fuel lines and the electrical harness is the same as used on similar engines. Attempts have been made to minimize the number of fluid connectors and to size the lines for flow velocities that would minimize system pressure drops. Where lube oil flows through areas of high temperature, special attention has been to minimize any traps that could lead to oil coking after engine shutdown.

Figure 46 shows how the various lines are directed through the flowpath strut and are collected in the isolation box. This isolation box prevents leakage of cooling air used for the inner core cowl area out of the flowpath strut. This figure also shows how the lines are bundled together and supported as they penetrate the strut.

The core compartment partition wall shown in Figure 47 provides a boundary for the higher pressure inner core cowl area from the area around the high pressure and low pressure turbine outer casings. An area of lower pressure around the turbine casing is required to provide a maximum ΔP for the turbine clearance control cooling airflow. The maximum pressure in the inner core cowl area is 26.2 KPa (3.8 psi) higher than the cavity around the turbine outer casings. This ΔP along with the temperature gradient of 315.6° C (600° F) along the wall (shown in Figure 47) was the predominant criteria used to determine the configuration of the partition wall. The calculated stresses are below the allowable stress of the Inco 625 material used.

The pipes that penetrate the partition are provided with sliding seals as shown in Figure 47. Shown in this figure is a metallic, spherical-type seal but a bellows seal is also being considered.

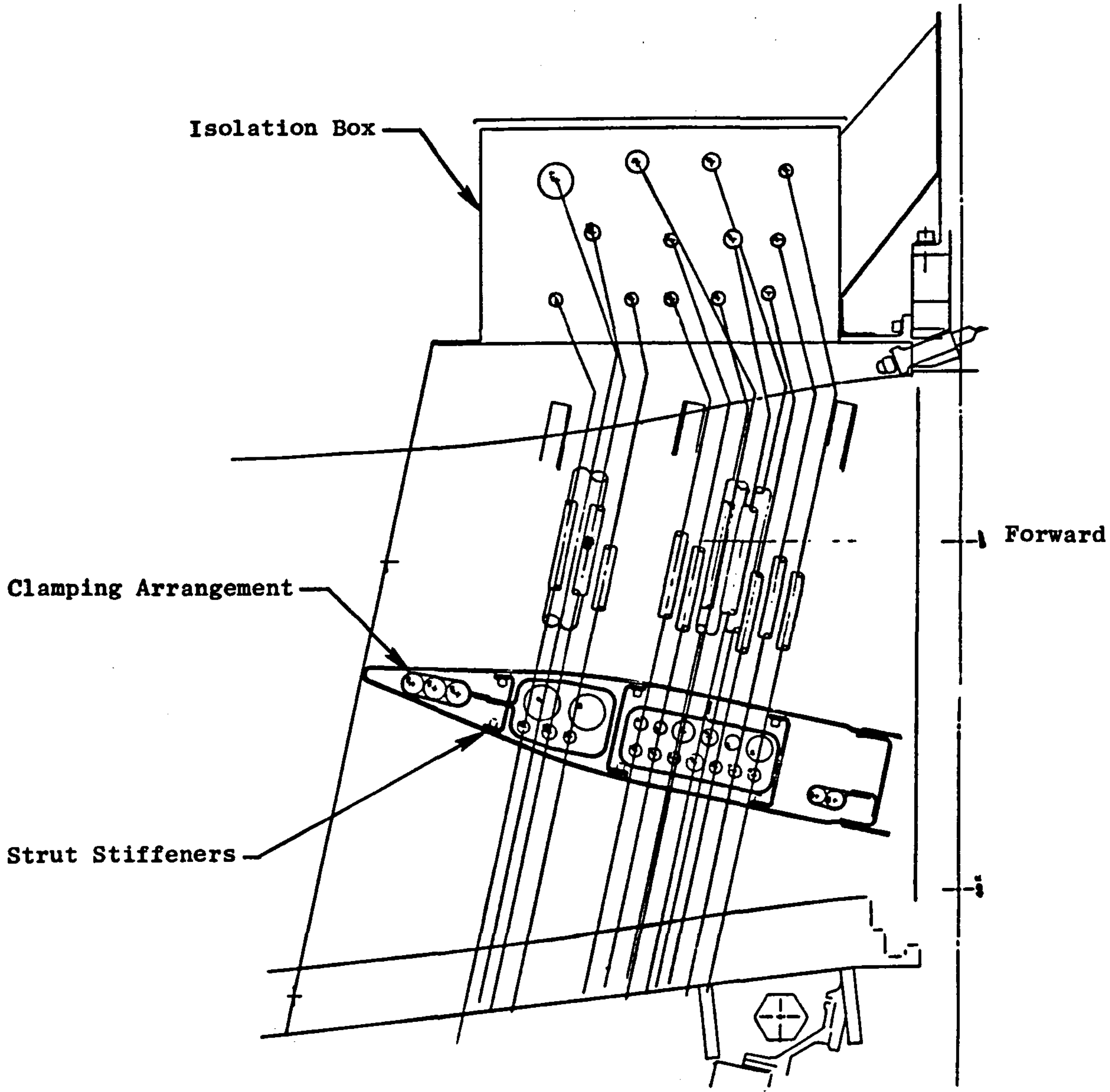


Figure 46. Configuration Design Isolation Box.

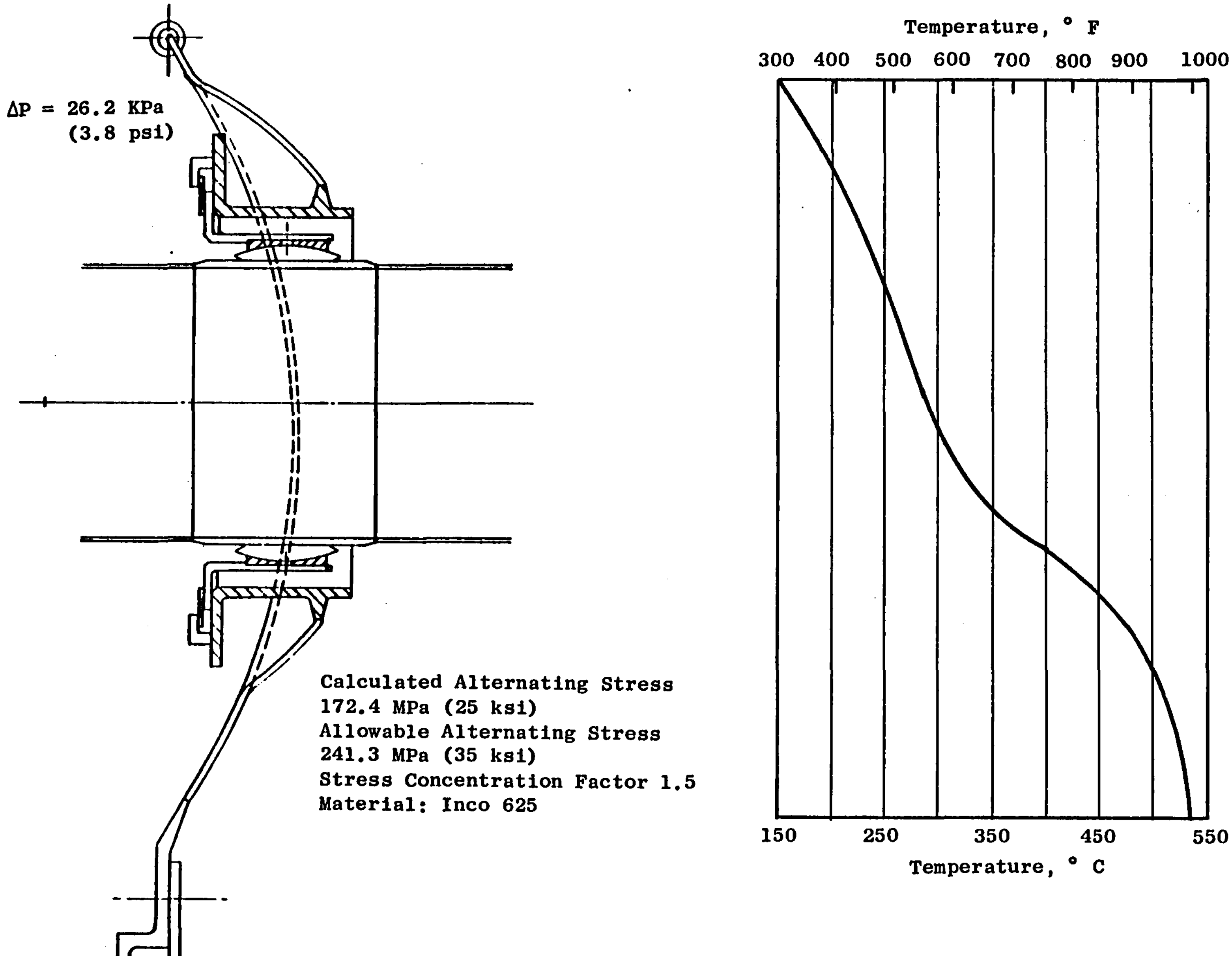


Figure 47. Configuration Design; Core Compartment Partition Wall.

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APPENDIX

LIST OF SYMBOLS AND NOMENCLATURE

AGB	-	Accessory Gearbox
BIR	-	Bearing Inner Race
CCW	-	Counterclockwise
CDP	-	Compressor Discharge Pressure
CML	-	Cubic Mean Load
DN	-	Bearing Design Parameter (Bore in mm x rpm)
E^3	-	Energy Efficient Engine
fpm	-	Feet Per Minute
FPS	-	Flight Propulsion System
g-in.	-	Gram Inch
g-mm	-	Gram Millimeter
gpm	-	Gallons Per Minute
HPT	-	High Pressure Turbine
ICLS	-	Integrated Core Low Spool (Turbofan Test Engine)
IRC	-	Internal Radial Clearance
kg/sec	-	Kilograms Per Second
KPa	-	Kilo Pascal
ksi	-	Thousand Pounds Per Square Inch
kW	-	Kilowatt
LCF	-	Low Cycle Fatigue
lpm	-	Liters Per Minute
LPT	-	Low Pressure Turbine
mils	-	1/100 Inch
MPa	-	Mega Pascal

N	-	Newton
Pa	-	Pascal
psi	-	Pounds Per Square Inch
psia	-	Pounds Per Square Inch Area
psid	-	Pounds Per Square Inch Diameter
PTO	-	Power Takeoff
T	-	Teeth
TF39-4B	-	General Electric Turbofan Engine
T/O	-	Takeoff
μ	-	Coefficient of Friction
β	-	Contact Angle
ϕ	-	Diameter

End of Document